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**AUTOMOTIVE TECHNOLOGY STATUS AND
PROJECTIONS VOLUME II. ASSESSMENT
REPORT**

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Automotive Technology Status and Projections Volume II. Assessment Report

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ABSTRACT

Results of an assessment of the status and projections of automotive technology are presented. Factors considered include fuel economy, exhaust emissions, multifuel capability, advanced materials, and cost/manufacturability for both conventional and advanced alternative power systems.

To insure valid comparisons of vehicles with alternative power systems, the concept of an Otto-Engine-Equivalent (OEE) vehicle was utilized. Each engine type was sized to provide equivalent vehicle performance. Sensitivity to different performance criteria was evaluated. Fuel economy projections are made for each engine type considering both the legislated emissions standards (0.4 g/mi HC, 3.4 g/mi CO, 1.0 g/mi NO_x) and possible future emissions requirements (0.4 g/mi NO_x).

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SECTION 1

INTRODUCTION

The need to improve air quality has led to more stringent Federal emissions standards for controlling exhaust emissions from automobiles. Early attempts by automobile manufacturers to meet these emissions standards with conventional engines led to significant fuel economy penalties. In 1975, when much emphasis was being placed on developing better emissions control hardware, the Jet Propulsion Laboratory published a report on an Automotive Power Systems Evaluation Study (APSES). This report made an assessment of conventional and alternative power systems and included projections of their fuel economy potential and their ability to meet the most stringent legislated emissions standards (0.4 g/mi HC, 3.4 g/mi CO, 0.4 g/mi NO_x).

Since the APSES report, the impact of the oil embargo by the OPEC nations has helped focus the need for energy conservation in this country, especially in the use of petroleum for transportation. This need to conserve petroleum has led to the passage of Federal fuel economy standards which require an increase from a sales-weighted average of 18 mpg in 1978 to a sales-weighted average of 27.5 mpg in 1985 based on the composite driving cycle. The passage of these Federal fuel economy standards has had and will continue to have a significant effect on the type of vehicles produced by the automotive industry. It is leading to the development of lighter weight vehicles and more fuel-efficient conventional engines.

The energy crisis and the fuel economy legislation have resulted in modification of emissions standards for passenger cars. Up through 1985, the most stringent Federal emissions standards are set at (0.4 g/mi HC, 3.4 g/mi CO, 1.0 g/mi NO_x). Because of its special pollution problems, California requires that vehicles meet the (0.4 g/mi HC, 9.0 g/mi CO, 0.4 g/mi NO_x) values starting in 1982. Consideration is being given to waiving the 0.4 g/mi NO_x requirement and requiring vehicles to be certified for 100,000 miles (currently a certification for 50,000 miles is required). The 0.4 g/mi NO_x requirement has been established as a research goal by Federal legislation.

The Department of Energy (DOE) has significant programs aimed at the development of advanced alternative heat engine power systems (Brayton and Stirling). Since these advanced power systems are aimed for automotive application in the 1985-2000 period, the DOE programs have as their prime goals good energy efficiency, multifuel capability, low emissions, and competitive cost.

Because of the ever-changing automotive technology base and the evolving constraints (fuel economy, exhaust emissions, noise, etc.) on the automotive industry, it is important to continually reassess the potential of the various alternative power systems in this changing environment. The objective of the work in the Automotive Technology Status and Projections (ATSP) task is to provide DOE with this information so that it can be used to help assess and develop a more effective DOE heat engine program in the transportation area.

The general objective of the ATSP task is to make a continuing assessment of the status of automotive technology. The study considers both current and advanced conventional engines, advanced alternative engines, advanced power train components and other energy-conserving vehicle modifications which could be implemented by the end of this century. A secondary objective is to make vehicle-level projections of the fuel economy and emissions potential of both conventional and advanced alternative power systems.

The major thrust of the ATSP task is the assessment of the potential of advanced alternative heat engine power systems (Brayton and Stirling) when compared with the evolving conventional power systems (uniform-charge Otto, diesel, stratified-charge Otto). Factors considered in the task include fuel economy, exhaust emissions, multifuel capability, use of advanced materials, and cost/manufacturability.

Since the period covered in the study extends to the end of the century, each candidate power system is evaluated for its ability to meet not only the current legislated emissions standards but also the possible future emissions requirements, including the 0.4 g/mi NO_x level. Consideration is given to the presently unregulated emissions of particulates, noise, and odor. The impacts of the Federal fuel economy standards are included as they relate to vehicle/engine size and weight.

To obtain the latest data and to learn about the latest developments in conventional and alternative power systems and advanced materials, technical discussions have been held with both domestic and foreign automobile manufacturers, automobile suppliers, and Government laboratories. In general, the technical material contained in this report is based on the information which was available as of April 1978.

SECTION 2

CONVENTIONAL POWER SYSTEMS

2.1 INTRODUCTION

The characteristics of both the vehicles and the conventional engines which will be marketed in 1985-90 are likely to be considerably different from those which were envisioned in 1970-73, when the alternative engine (gas turbine and Stirling) programs were initiated, or even in 1974, when the comprehensive assessment of the potential of those alternative engines was completed by the Jet Propulsion Laboratory (Reference 2-1). Hence, in setting goals for the alternative engine R&D programs and assessing their long-term potential advantages, it is necessary to consider the changes in conventional automotive technology and the effect of these changes on the so-called moving baseline with which the alternative engines will inevitably be compared. In this section, recent trends in vehicle design and conventional engine and transmission development are summarized and used as the base for determining a moving baseline for the period 1975-85. The moving baseline, utilizing conventional engine systems, is given in terms of engine weight and size, and vehicle fuel economy and energy intensity (Btu/veh-mi), for both the EPA city and highway cycles. The moving baseline reflects changes in both vehicle and engine design which have occurred and will continue to occur in response to the Federal emissions and fuel economy standards summarized in Tables 2-1 and 2-2.

The general approach taken in developing the baseline for 1975-85 was to correlate, as a function of engine type and vehicle inertia weight, engine and vehicle technical data obtained from the automobile industries of the United States, Europe, and Japan with vehicle emissions and fuel economy data for the 1973-78 period published by EPA. The conventional engine types considered in this study are (1) the uniform-charge gasoline engine with and without turbocharging, (2) the diesel engine with and without turbocharging, (3) the rotary engine, and (4) the three-valve, stratified-charge gasoline engine. As discussed in the following sections, significant technical developments in conventional engines have resulted in both lower pollutant emissions and increased fuel economy. In addition, the general downsizing of passenger cars to meet the mandated corporate fuel economy standards has resulted in reduced horsepower requirements and created a need for improved engine specific weight and volume.

Updated projections of vehicle characteristics and engine requirements for 1985 and the baseline fuel economy and energy intensity of those vehicles using advanced conventional engines (gasoline and diesel fueled) are presented in later sections. These projections represent an updated moving baseline for conventional engine-powered passenger cars and, as such, can be used in setting goals for and assessing the progress in alternative engine R&D programs.

Table 2-1. Emission Standards for Light-Duty Vehicles

Year	Emission Standards, g/mi					
	California			Other 49 States and Federal		
	HC	CO	NO _x	HC	CO	NO _x
1973 ⁽¹⁾	3.2	39	3	3.2	39	3
1974	3.2	39	3	3.2	29	3
1975 ⁽²⁾	0.9	9	2	1.5	15	3
1976	0.9	9	2	1.5	15	3
1977	0.4	9	1.5	1.5	15	2
1978	0.4	9	1.5	1.5	15	2
1979	0.4	9	1.5	1.5	15	2
1980	0.4	9	1.0	0.4	15	2
1981	0.4	9	1.0	0.4	7	1 ⁽³⁾
1982	0.4	9 ⁽⁴⁾	0.4 ⁽⁵⁾	0.4	3.4	1
1983	0.4	9	0.4	0.4	3.4	1
1984	0.4	9	0.4	0.4	3.4	1
1985	0.4	9	0.4	0.4	3.4	1

(1) 1972 CVS-C test procedures used for 1973-74.

(2) 1975 CVS-CH test procedure used for 1975 and beyond.

(3) Diesels and cars with other innovative fuel saving engines could qualify for a NO_x standard of 1.5 g/mi (1977 amendments to the 1970 Clean Air Act).

(4) California is considering a CO standard of 7 g/mi.

(5) California is considering a NO_x standard of 1 g/mi if vehicle can be certified for 100,000 mi rather than 50,000 mi.

Table 2-2. Mandatory Fuel Economy Standards

Year	Sales Weighted Average, mpg ⁽¹⁾
1978	18
1979	19
1980	20
1981	22
1982	24
1983	26
1984	27
1985	27.5

(1) Composite - 55% urban cycle, 45% highway cycle

2.2 RECENT TRENDS IN VEHICLE DESIGN AND CONVENTIONAL ENGINE DEVELOPMENT

Recent trends in vehicle design and conventional engine/transmission development which impact the moving baseline determination are listed below.

2.2.1 Vehicle Design Trends

- (1) Downsizing - reduction of exterior dimensions while maintaining essentially the same interior dimensions.
- (2) Weight reduction by material substitution and downsizing.
- (3) Increased use of front-wheel drive.
- (4) Use of lower power-to-weight ratio - reduced performance.

2.2.2 Engine/Transmission Trends

- (1) Development of more compact, lighter, higher rpm, four- and six-cylinder gasoline engines (in-line and V configurations).
- (2) Development of three-way catalyst emission control systems and their limited use in production vehicles.
- (3) Turbocharging of several production gasoline engines.
- (4) Development of several passenger car diesel engines by the redesign of production gasoline engines.
- (5) Turbocharging of diesel engines for passenger cars.
- (6) Refinement of the three-valve, stratified-charge gasoline engine.
- (7) Significant improvements in the fuel consumption of rotary engines using both the uniform- and stratified-charge approaches.
- (8) Increased availability of four- and five-speed manual transmissions.
- (9) Introduction of automatic transmissions with torque converter lockup.

The way in which each of these vehicle/engine developments impacts the moving baseline is discussed quantitatively in subsequent sections.

2.3 CHARACTERIZATION OF ENGINE, TRANSMISSION, AND VEHICLE DESIGNS

2.3.1 Engines and Transmissions

Passenger Car Engines. The weight and size of various types of conventional engines which are currently used or being developed for use in passenger cars are listed in Table 2-3. The data shown have been collected from various sources, as indicated by References 2-2 through 2-6 and private communications from Toyota, Ford, and Borg-Warner. The specific weights (hp/lb) of the European-type, naturally aspirated four-cylinder gasoline engines range from 0.3 to 0.35; those for the turbocharged versions are about 0.40. Naturally-aspirated diesel engines of recent design have a specific weight of 0.15 to 0.2 hp/lb; values for the turbocharged versions are about 0.25.

Table 2-3. Engine Size and Weight Characteristics

Model	Type	hp	hp/lb	hp/CID	Engine Dimensions, in.		
					L	W	H
SMALL ENGINES							
Honda/Accord	CVCC-L4	67	0.17	0.68	22	21	23
Ford/Fiesta	Gas-L4	66	0.34	0.68	17	13	19
VW/Rabbit	Gas-L4	78	0.28	0.80	17	17	22
Toyota	Gas-L4	58	0.24	0.82			
	Gas-L4	75	0.23	0.77	22	21	23
	Gas-L4	95	0.23	0.71			
VW/Rabbit	Diesel-L4	50	0.18	0.55	17	17	22
	Diesel-L4 (TC)	69	0.24	0.76	17	17	22
Ford/Pinto	Gas-L4	88	0.23	0.51	31	20	27
Mercedes-Benz	Diesel-L4	62	0.14	0.45			
LARGER ENGINES							
Peugeot	Gas-V6	130	0.33	0.80	19	25	25
Saab	Gas-L4	110	0.32	0.91	20	20	26
	Gas-L4(TC)	125	0.39	1.1	20	20	26
Ford	Gas-V8	145	0.22	0.41	30	30	30
Buick	Gas-V6	105	0.22	0.45			
	Gas-V6(TC)	150	0.32	0.65			
Ford	Gas-L6	85	0.17	0.43	30	16	26
NSU	Rotary-2 rotor (45 CID/rotor)	170	0.43	0.98	18	29	23
VW	Diesel-L6	78	0.192	0.59	31	19	31
	Diesel-L6(TC)	105	0.25	0.67	31	19	31
Mercedes-Benz	Diesel-L5	80	0.15	0.42			
	Diesel-L5(TC)	105	0.20	0.55			
Oldsmobile	Diesel-V8	120	0.16	0.34	32	28	30

The weight characteristics for several conventional engines (uniform-charge gasoline, naturally-aspirated diesel, turbocharged diesel) are given in Figures 2-1, 2-2, and 2-3. For conventional engines of a given type, there are considerable differences in engine weights, as is shown by the data bands. The lighter weight engines are representative of the more recent designs. The solid lines through the data were used to describe those engine types in the comparisons discussed in Section 9. Weight characteristics for several advanced conventional gasoline engines (rotary uniform-charge, stratified-charge, rotary stratified-charge) are given in Figures 2-4, 2-5, and 2-6. In this case, much less data is available for establishing representative weight characteristics. The attractive weight characteristics of the rotary engine are clearly shown in these figures.

The shape of the engine torque-speed characteristic is important in determining the acceleration capability of a vehicle. Torque-speed data for several conventional engines are given in Figures 2-7, 2-8, and 2-9. The engine torque and speed are normalized with respect to the torque and speed at maximum power. The solid lines represent the torque-speed characteristics used in the comparisons in Section 9. Torque-speed data for several advanced conventional gasoline engines are shown in Figures 2-10, 2-11, and 2-12.

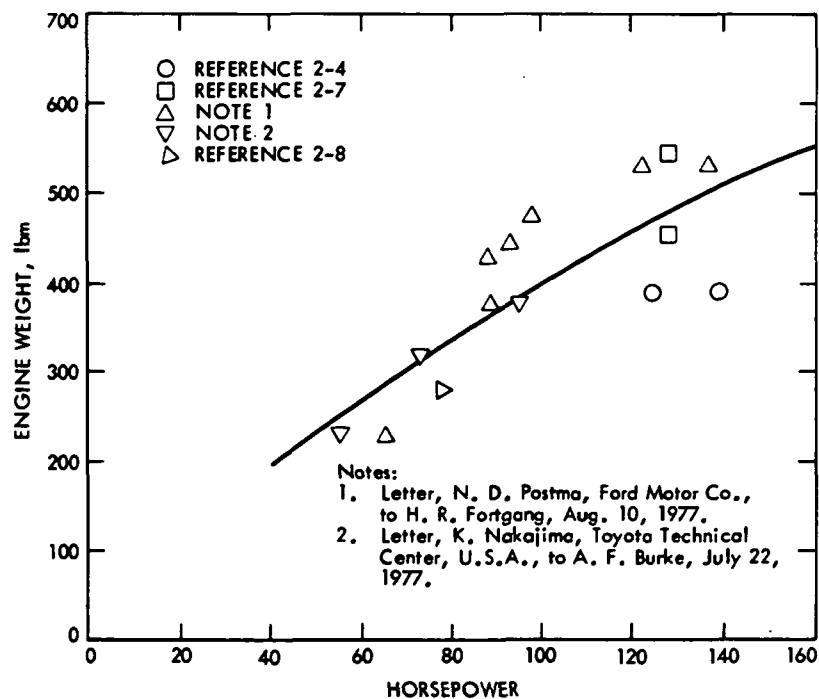


Figure 2-1. Engine Weights for Uniform-Charge Otto

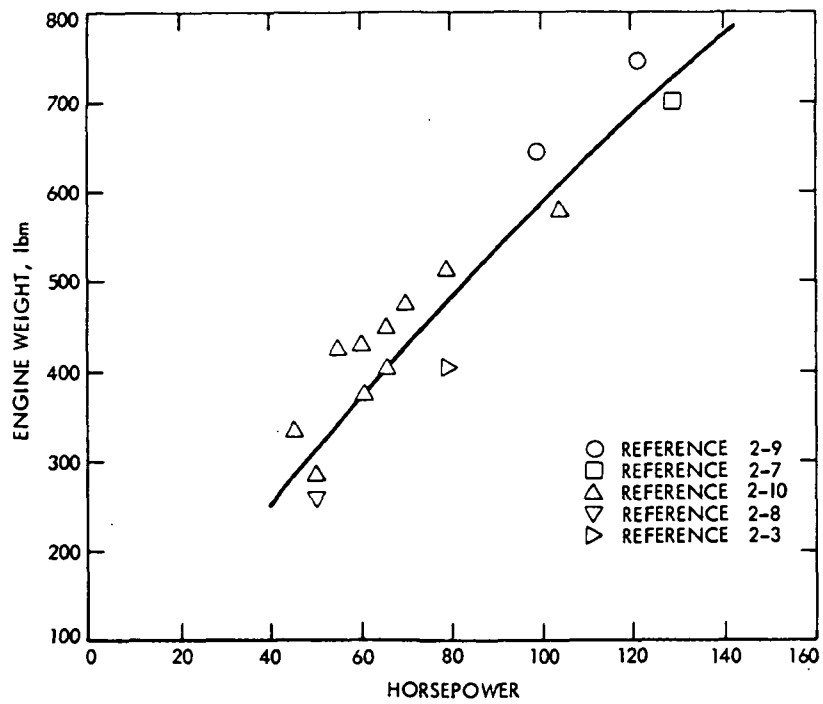


Figure 2-2. Engine Weights for Naturally-Aspirated Diesel

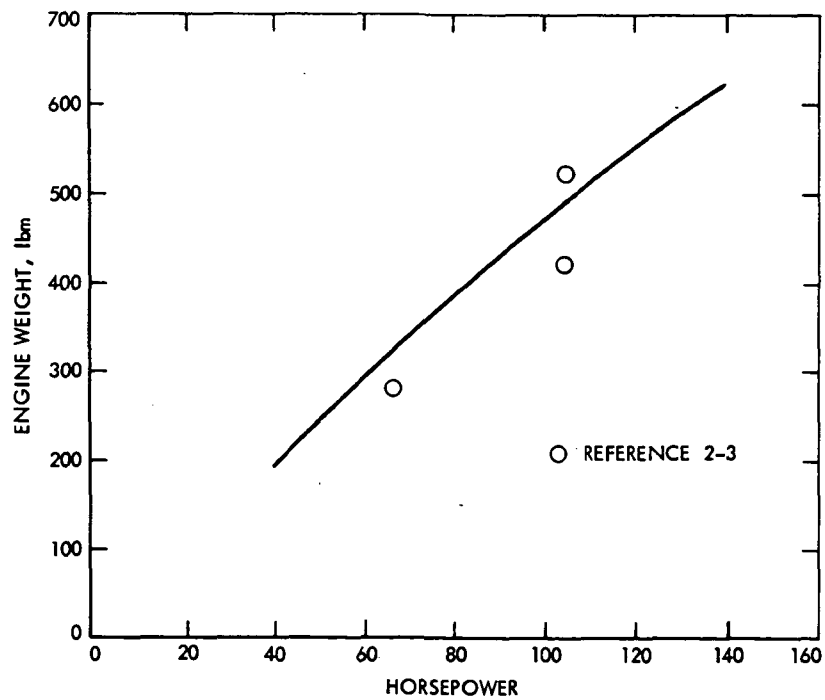


Figure 2-3. Engine Weights for Turbocharged Diesel

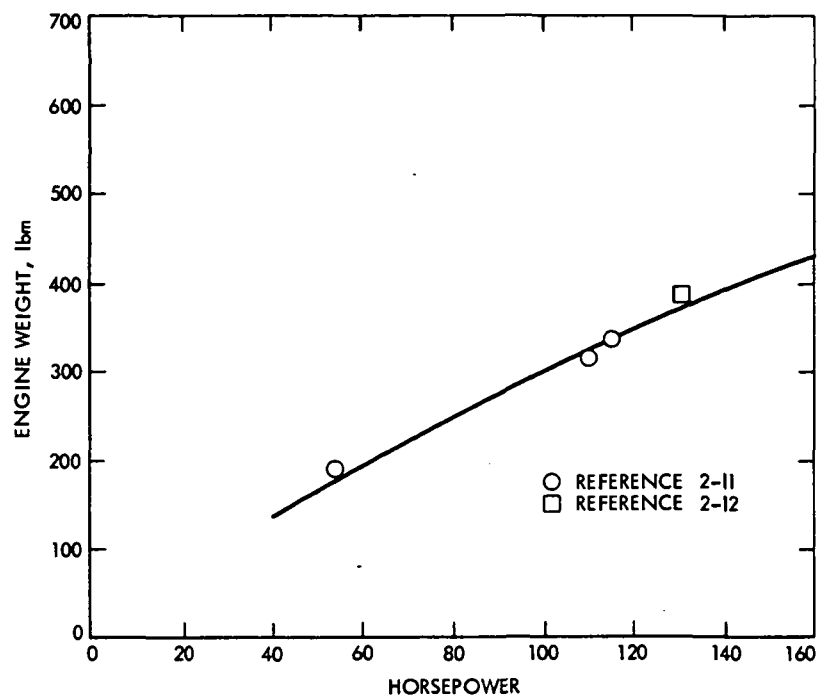


Figure 2-4. Engine Weights for Uniform-Charge Otto (Rotary)

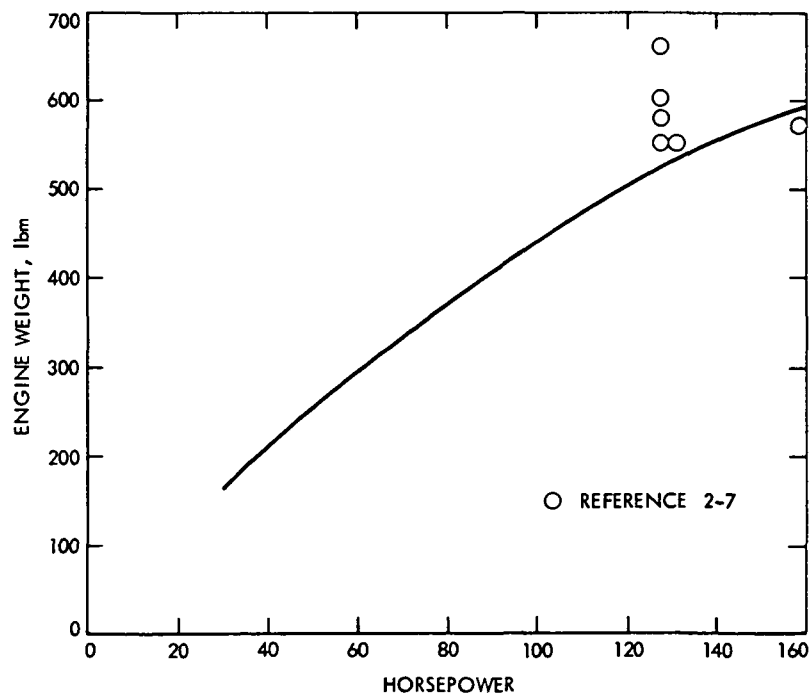


Figure 2-5. Engine Weights for Stratified-Charge Otto

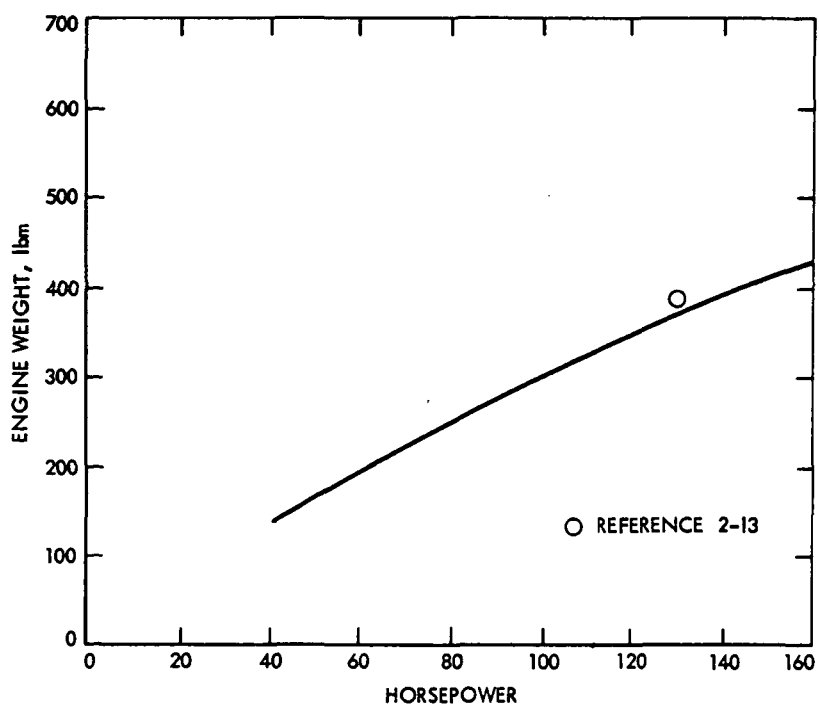


Figure 2-6. Engine Weights for Stratified-Charge Otto (Rotary)

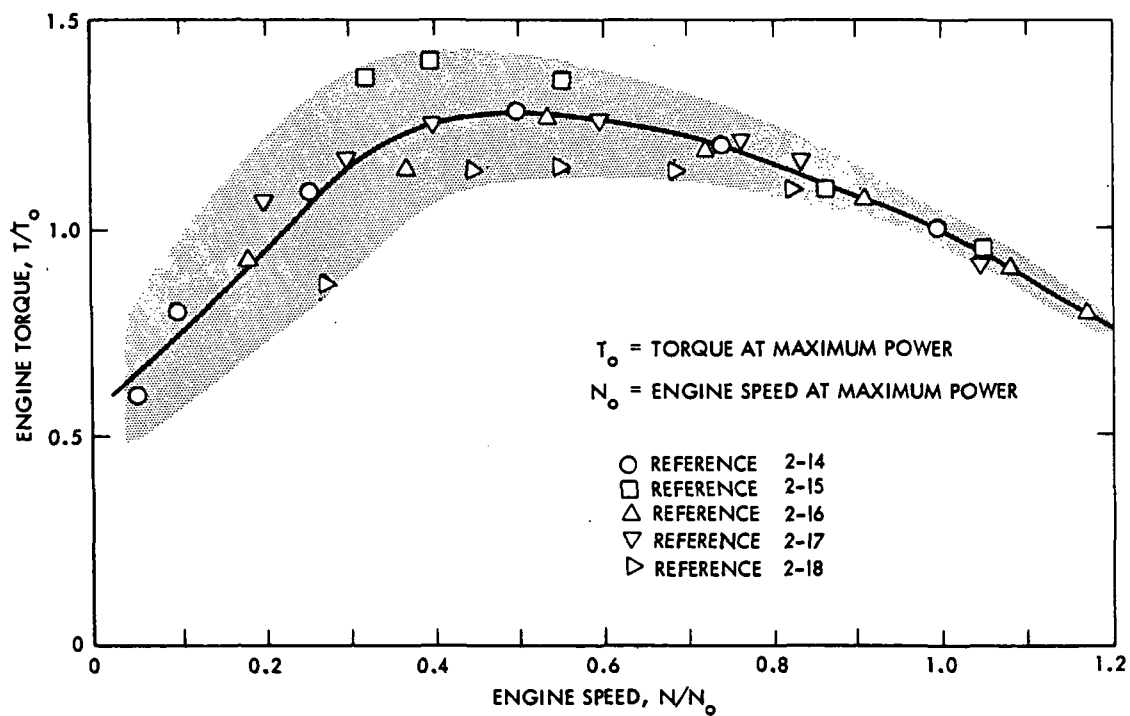


Figure 2-7. Torque-Speed Characteristics for Uniform-Charge Otto

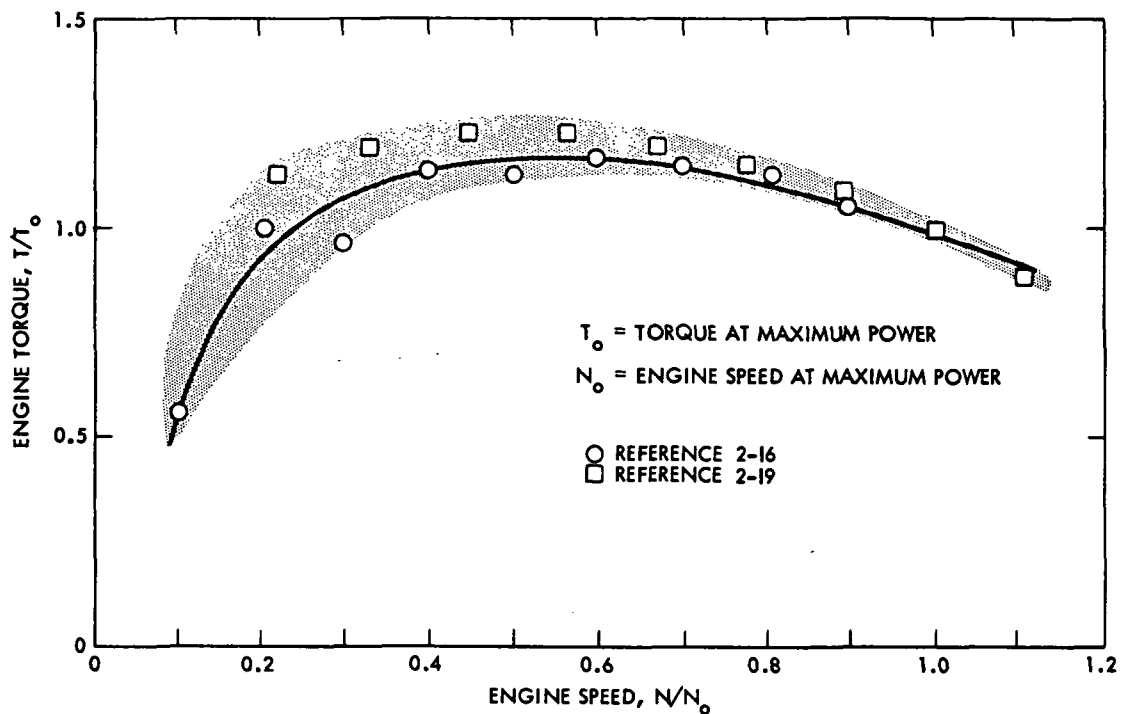


Figure 2-8. Torque-Speed Characteristics for Naturally-Aspirated Diesel

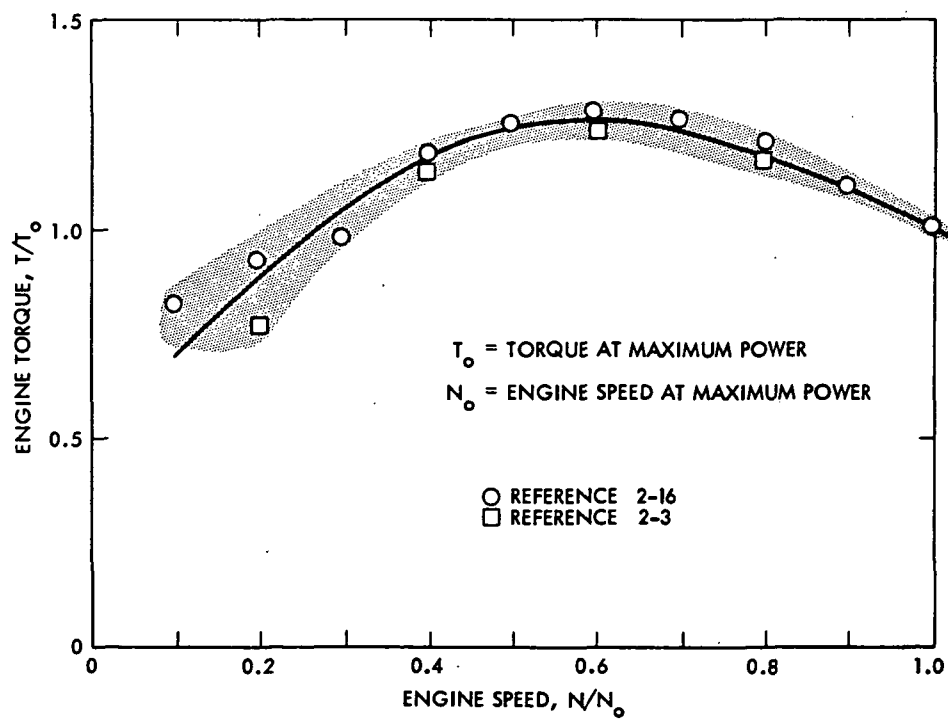


Figure 2-9. Torque-Speed Characteristics for Turbocharged Diesel

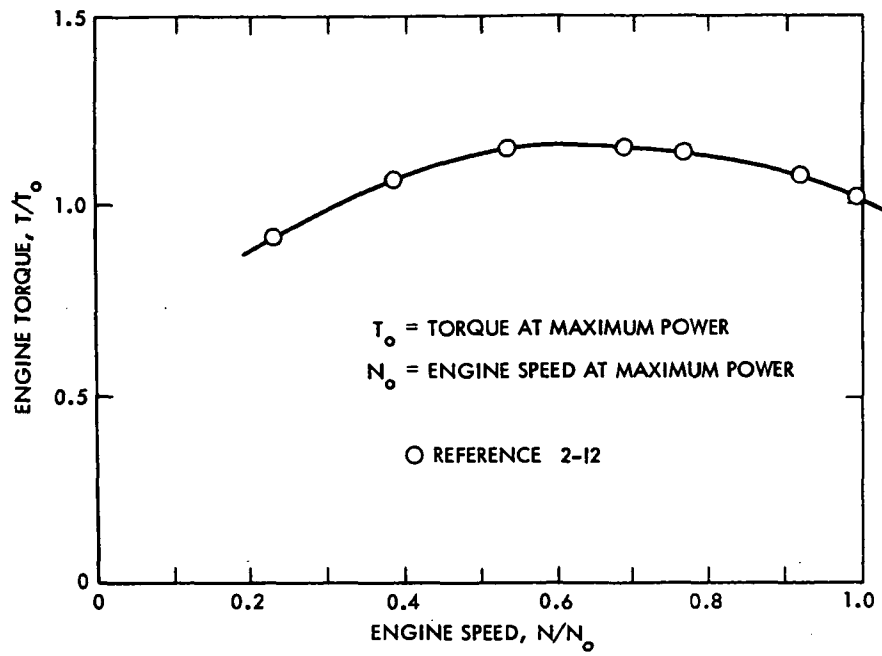


Figure 2-10. Torque-Speed Characteristics for Uniform-Charge Otto (Rotary)

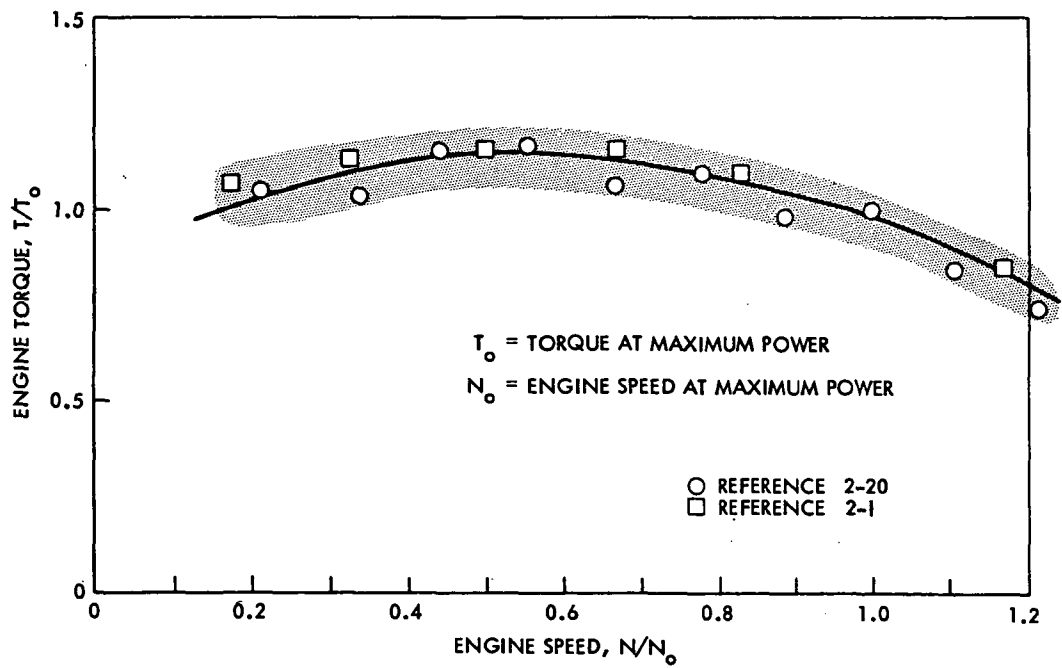


Figure 2-11. Torque-Speed Characteristics for Stratified-Charge Otto

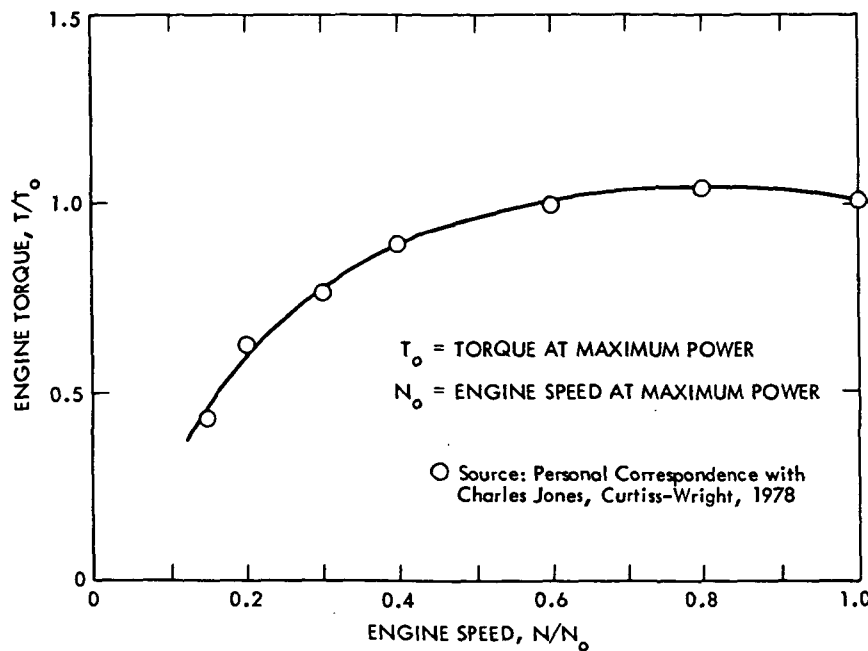


Figure 2-12. Torque-Speed Characteristics for Stratified-Charge Otto (Rotary)

The fuel consumption and emissions characteristics of the various engines are most easily seen from the EPA emissions and fuel economy results (Reference 2-21) for various engine/transmission/vehicle combinations. Summaries of fuel economy data for 1973 to 1978 for conventional four-, six-, and eight-cylinder engines used in domestic passenger cars are given in Table 2-4. Corresponding summaries for the Honda CVCC and Mazda rotary engines are given in Tables 2-5 and 2-6. The data presented in the tables indicate that fuel economy (for cars of fixed inertia weight) improved steadily from 1973 to 1978, even though the emission standards became progressively stricter during that period. This was possible because of oxidizing catalysts, introduced on most cars in 1975-76, and the use of the three-way (reducing/oxidizing) catalysts on some California cars in 1977-78.

Fuel economy and emissions data for 1976-78 production vehicles and selected prototype vehicles using advanced conventional engine systems are given in Table 2-7. The table includes data for gasoline engines using the three-way catalyst and turbocharging, and diesel engines using EGR and turbocharging. It seems reasonable to conclude from the data given in Table 2-7 that, with further development of existing technology, the 1984 Federal emissions standards (0.4 g/mi HC, 3.4 g/mi CO, 1.0 g/mi NO_x) can be met in either gasoline- or diesel-powered (naturally aspirated or turbocharged) vehicles with inertia weights up to 4000 lb. The Volvo/Saab results for L4 and V6 fuel-injected gasoline engines using a three-way catalyst indicate that the 1982

Table 2-4. Historical Review (1973-78) of Fuel Economy
of Domestic Passenger Cars Using
Conventional Engines

Car Model	Year	Inertia Weight, lb	Trans- mission	Fuel Economy, mpg	
				City	Highway
ENGINE: FORD L4-140 CID					
Capri (122 CID)---	1973	2750	A3	17.5	
Pinto-----	1974	2750	A3	16.7	
Pinto-----	1975	2750	A3	18	26
Pinto-----	1976	2750	A3	21	31
	C-76 ⁽¹⁾	2750	A3	21	30.5
Pinto-----	1977	2750	A3	22.5	32
	1977	2750	M4	24.2	36.7
	C-77	2750	A3	22	30
	C-77	2750	M4	23	34
Pinto-----	1978	2750	M4	25	35
	C-78 ⁽²⁾	2750	M4	22	31
ENGINE: GM L4-140/151 CID					
Vega-----	1973	2750	A3	18.9	
Vega-----	1974	3000	A3	20.1	
Vega-----	1975	3000	A3	21	29
Vega-----	1976	3000	A3	20	29
	C-76	3000	A3	18.5	27.5
Vega-----	1977	3000	A3	21	29
	C-77	3000	A3	21	30
Astre (151 CID)---	1977	3000	A3	23	31
	C-77	3000	A3	20	27
Sunbird (151 CID)-	1978	3000	A3	23	31
	C-78 ⁽²⁾	3000	A3	23	31
ENGINE: GM L6-250 CID					
Chevelle-----	1973	3500	A3	12.8	
Nova-----	1974	3500	A3	15.7	
Nova-----	1975	4000	A3	16	22
Nova-----	1976	3500	A3	18	24
	C-76	3500	A3	15	21
Nova-----	1977	3500	A3	19	24
	C-77	3500	A3	15	22
Nova-----	1978	3500	A3	18	24

(1) "C" refers to cars built to meet stricter California standards.

(2) Equipped with a three-way catalyst.

Table 2-4. Historical Review (1973-78) of Fuel Economy of Domestic Passenger Cars Using Conventional Engines (Continuation 1)

Car Model	Year	Inertia Weight, lb	Trans- mission	Fuel Economy, mpg	
				City	Highway
ENGINE: CHRYSLER L6-225 CID					
Plymouth-----	1973	3500	A3	15.5	
Plymouth-----	1974	3500	A3	16.5	
Dart-----	1975	3500	A3	18	23
Dart-----	1976	3500	A3	18.5	23.5
	C-76	3500	A3	14.5	20
Aspen-----	1977	3500	A3	19.5	26
	C-77	4000	A3	16	19
Aspen-----	1978	3500	A3	20	27
ENGINE: FORD L6-250 CID					
Mustang-----	1973	3000	A3	12.7	
Torino-----	1974	3500	A3	14.0	
Maverick-----	1975	3500	A3	15.5	20.5
Maverick-----	1976	3500	A3	17	21
	C-76	3500	A3	15.5	21
Granada-----	1977	3500	A3	17.5	22.5
	C-77	3500	A3	18.0	23
Granada-----	1978	3500	A3	18	26
ENGINE: FORD V8-351 CID					
Torino-----	1973	4500	A3	9.0	
Torino-----	1974	4500	A3	9.9	
Torino-----	1975	4500	A3	11.5	16.0
LTD II-----	1976	4500	A3	13.5	18.5
	C-76	4000	A3	12.0	17.0
LTD II-----	1977	4500	A3	13.5	19.5
	C-77	4500	A3	12.0	18.0
Thunderbird-----	1978	4500	A3	15	22
ENGINE: GM V8-350 CID					
Impala-----	1973	4500	A3	11.0	
Ventura-----	1974	4000	A4	9.5	
Nova-----	1975	4000	A3	14.0	19.0
Camaro-----	1976	4000	A3	14.0	18.5
	C-76	4000	A3	13.0	16.0
Nova-----	1977	4000	A3	14.5	19.0
	C-77	4000	A3	14.0	19.0
Impala-----	1978	4000	A3	15	21.5

Table 2-5. Historical Review (1974-77) of Fuel Economy of Passenger Cars Using Honda CVCC Engine

Car Model	Year	Inertia Weight, lb	Trans- mission	Fuel Economy, mpg	
				City	Highway
Prototype (76 CID)-	1974	2000	M4	25	35
Civic (76 CID)-----	1975	2000	M4	28	38
		2000	M5	28	42
Civic (76 CID)-----	1976	2000	M4	28	41
Civic (91 CID)-----	1976	2000	M5	31	44
	C-76	2000	M4	32	42
	C-76	2000	M5	31	44
Civic (76 CID)-----	1977	2000	M4	28	43
Civic (91 CID)-----	1977	2000	M4	39	50
	C-77	2000	M4	35	46
Accord (98 CID)-----	1977	2250	M5	38	48
	C-77	2250	M5	33	47

Table 2-6. Historical Review (1973-78) of Fuel Economy of Passenger Cars Using Mazda Rotary RX-3 (70 CID) Engine

Car Model	Year	Inertia Weight, lb	Trans- mission	Fuel Economy, mpg	
				City	Highway
RX-2 Coupe-----	1973	2750	M4	12.3	
RX-3 Coupe-----	1974	2750	M4	13.3	20
	1975		Not available		
RX-3 Coupe-----	1976	2750	M5	17	25
(120 hp)	C-76	2750	M5	19	29
RX-3 Coupe-----	1977	2750	M4	18	26
(120 hp)		2750	M5	18	29
	C-77	2750	M4	18	26
		2750	M5	18	29
RX-3 Coupe-----	1978	2750	M5	19	28
(120 hp)					

California standard (0.4 g/mi HC, 9.0 g/mi CO, 0.4 g/mi NO_x) can be met using such a system for 3500/4000-lb inertia weight cars. It is not clear, however, whether or not carbureted engines in similar size cars will be able to meet the 1982 California standard with or without a significant fuel economy penalty. There are no currently available data to suggest

Table 2-7. Fuel Economy and Emissions of Vehicles Having Advanced Conventional Engine Systems

Mfgr.	Model	Inertia Weight, lb	Emission Standard(1)	Engine(2)	Emission Control System	Emissions, g/mi			Fuel Economy, mpg	
						HC	CO	NO _x	City	Highway
Saab	99	3000	C-78	Gasoline, L4 121 CID, TC	FI, 3-way catalyst	0.26	2.9	1.1	19.7	27.2
	99	3000	C-78	Gasoline, L4 121 CID, NA	FI, 3-way catalyst	0.19	3.9	0.28	22.2	29.7
	99	3000	F-77	Gasoline, L4 121 CID, NA	FI, air inj., EGR	0.89	8.0	1.7	21.9	31.0
Volvo	240	3000	C-77	Gasoline, L4 130 CID	FI, 3-way catalyst	0.2	2.7	0.18	18	28
	265	3500	C-77	Gasoline, V6 163 CID	FI, oxid. cat., EGR	0.2	1.1	1.25	14	26
	244	3500	C-78	Gasoline, V6 163 CID	FI, 3-way catalyst	0.35	3.6	0.85	15.6	26
	264	3500	F-76	Gasoline, V6 163 CID	FI, air inj., EGR	1.3	13.0	2.6	15.0	27
	244	3500	F-76	Gasoline, L4 130 CID	FI, air inj., EGR	0.80	9.1	1.9	18	25
GM	Sunbird	3000	C-77	Gasoline, L4 151 CID, A3	Carb., oxid. cat., EGR	0.25	2.5	1.3	20	27
	Sunbird	3000	C-78	Gasoline, L4 151 CID, A3	Carb., 3-way cat.	0.38	5.7	1.2	23	31
	Sunbird	3000	F-77	Gasoline, L4 151 CID, A3	Carb., oxid. cat., EGR	0.5	7.0	1.1	22.5	30.5

Table 2-7. Fuel Economy and Emissions of Vehicles Having Advanced Conventional Engine Systems (Continuation 1)

Mfr.	Model	Inertia Weight, lb	Emission Standard(1)	Engine(2)	Emission Control System	Emissions, g/mi			Fuel Economy, mpg	
						HC	CO	NO _x	City	Highway
Ford	Pinto	2750	C-77	Gasoline, L4 140 CID	Carb., oxid. cat., EGR	0.22	1.1	0.75	22	34
	Pinto	2750	C-78	Gasoline, L4 140 CID	Carb., 3-2ay catalyst	0.39	3.8	0.90	22.6	31
	Pinto	2750	F-77	Gasoline, L4 140 CID	Carb., oxid. cat., EGR	0.70	5.5	1.4	24	37
Peugeot	504D	3500	F-77	Diesel, L4 141 CID	None	0.65	1.6	1.1	28	34
	240D	3500	F-77	Diesel, L4 147 CID	None	0.35	1.1	1.5	25	34
Mercedes-Benz	300D	4000	F-77	Diesel, L5 183 CID	None	0.1	1.0	1.7	23	28
	Rabbit	2250	F-78	Diesel, L4 90 CID	None	0.3	1	1.05	40	53
VW	Dasher	2500	F-77	Diesel, L4 90 CID	None	0.5	1	1.15	35	47
	Rabbit Prototype	2250	----	Diesel, L4 90 CID, TC	None	0.22	0.98	0.93	42 (47.3 composite)	56 (42 composite)
		2750		Diesel, L4 90 CID, TC	None	0.15	0.7	1.1	(42 composite)	(42 composite)
		3000		Diesel, L4 90 CID, TC	None	0.14	0.8	1.2	(40 composite)	(40 composite)

Table 2-7. Fuel Economy and Emissions of Vehicles Having Advanced Conventional Engine Systems (Continuation 2)

Mfgr.	Model	Inertia Weight, lb	Emission Standard(1)	Engine (2)	Emission Control System	Emissions, g/mi				Fuel Economy, mpg	
						HC	CO	NO _x		City	Highway
VW(cont)		2250		Diesel, L4 90 CID, TC	EGR	0.2	1.2	0.4		(42 composite)	
		2750		Diesel, L4 90 CID, TC	EGR	0.17	1.5	0.4		(39 composite)	
		3000		Diesel, L4 90 CID, TC	EGR	0.17	1.6	0.50		(37 composite)	
	Rabbit VW/DOT Res. Veh.	2250	0.41/3.4/1.5	Diesel, L4, 40 CID, TC M5	None	---	---	---		55 (60 composite)	68
General Motors	Olds 88 & 98	4500	1.5/15/2	Diesel, V-8 350 CID, A3	None	1.0	2.0	1.6		20	27

(1) F = Federal emissions standards; C = California emissions standards.

(2) All vehicles equipped with M4 transmissions unless noted otherwise.

that 3500/4000-lb cars using diesel engines can meet the 0.4 g/mi NO_x standard; however, the emission results obtained by VW in the turbo-charged diesel Rabbit suggest that it may be possible to do so using EGR in small cars (2000/2500 lb). The limited data available indicate that, even for small cars, there is likely to be a 5-10% fuel economy penalty. In addition, engine smoking and durability may be a problem in using EGR with the diesel engine.

Transmissions. In the last several years, there have been significant changes in both the types and the characteristics of transmissions used in passenger cars. It is likely that even greater changes will occur in the future as the automobile industry seeks to increase vehicle fuel economy by reducing the weight of the transmission units, improving their mechanical efficiency, and operating the engine closer to its minimum bsfc condition. As indicated in Table 2-8, significant weight reductions can be achieved by simply fabricating the transmission housing from aluminum rather than cast iron. Permitting the engine to operate at lower bsfc and/or improving the transmission mechanical efficiency requires more profound changes in the design and characteristics of the transmission - for example, increasing the number of gears and overall gear ratio and locking up the torque converter when its fluid coupling function is not needed. The wide variety of manual and automatic transmissions now available is shown in Table 2-9. The first of the redesigned automatic transmissions using torque converter lockup was introduced by Chrysler in some of their 1978 models. Additional advanced transmissions utilizing four speeds (the fourth gear being overdrive) with lockup in third and fourth gears or a power split capability will be introduced by the automobile industry in the next 2 or 3 years.

The results of a recent computer study of various transmission options by GM are summarized in Table 2-10 (Reference 2-22). These results show the incremental improvement in fuel economy that can be expected by increasing the number of gears and overall gear speed ratio and utilizing torque converter lockup in one or more gears. The base-line cases were stock manual and automatic transmissions. The GM computer results represent differences between vehicle driveline systems which are optimized with respect to engine hp, axle ratio, and shift schedule. For that reason the results do not necessarily mean that the same fuel economy differences would be seen in practice in vehicles utilizing the different transmissions.

The GM study also considered the use of continuously variable transmissions (CVT) and found that the CVT yielded a fuel economy increase only a few percent better than a four-speed automatic with lockup in all gears. It was assumed in the study that the CVT and advanced automatic had the same overall speed ratio capability and that the automatic was shifted such that the transmission was always in the most efficient gear for the required road load. Increasing the overall speed ratio capability of the CVT and imposing a more realistic shift and lockup schedule on the automatic should significantly increase the

Table 2-8. Weights of Manual and Automatic Transmissions

Type	Rating, hp	Housing Materials	Weight, lb
M4	50-70	Aluminum	55
M4	90-100	Iron	97
		Aluminum	60
M5	90-100	Aluminum	70
M4	>150	Aluminum	70
A3	90	Iron	130
	130	Iron	155
	165	Iron	200

relative fuel economy advantage of the CVT over the advanced automatic. The introduction of a CVT is not assumed in the projections that follow. The introduction of improved automatic transmissions having performance comparable to the present four-speed manual (M4) transmission is, however, assumed.

EPA fuel economy data for a number of vehicles using various transmissions including the recently introduced Chrysler three-speed automatic (A3) transmission with lockup in third gear, are compared in Table 2-11. The data shown indicate that vehicles equipped with four-speed manual transmissions get on the average about 8% better fuel economy on the city cycle and about 15% better on the highway cycle than vehicles using standard three-speed automatics. The computer results given in Table 2-10 suggest that the fuel economy penalty associated with the use of an automatic transmission can be significantly reduced by incorporating overdrive and torque converter lockup. A preliminary analysis of the 1978 certification results by Chrysler of vehicles using their three-speed automatic with lockup indicates a 4% improvement in city and a 5% improvement in highway fuel economy for lockup at 25 mph. Greater improvements, especially on the highway cycle, can be expected as the transmission system is further developed and optimized.

Table 2-9. Characteristics of Manual and Automatic Transmissions
Used in Passenger Cars

Vehicle Mfgr.	Type	Gear Ratio for Gear No.					Torque Converter Ratio at Stall
		1	2	3	4	5	
GM	A3	2.52	1.52	1.0			2.25
	M3	3.11	1.84	1.0			
	M4	3.11	2.2	1.47	1.0		
	M5	3.40	2.08	1.39	1.0	0.8	
Ford	A3	2.46	1.46	1.0			2.01
	M3	3.56	1.90	1.0			
	M4	3.29	1.84	1.0	0.81		
Toyota	A3	2.45	1.45	1.0			2.2
	M4	3.58	2.08	1.4	1.0		
	M5	3.29	2.04	1.39	1.0	0.85	
VW	A3	2.65	1.59	1.0			2.1
	M4	3.45	1.95	1.37	0.97		
	M5	3.45	1.94	1.29	0.97	0.75	
Chrysler	A3	2.45	1.45	1.0 ⁽¹⁾			2.01
(1) Lockup in third gear at 20-30 mph.							

2.3.2 Vehicle Design

The vehicle designs will be discussed in terms of four classes - small (four passengers, two in front and two in rear with reduced comfort), compact (four passengers), mid-sized (five passengers), and full-sized (6 passengers). The primary distinguishing factor between these various classes is the interior size of the vehicle, and thus its capacity for carrying the specified number of passengers (adults) in comfort for a reasonable distance. The weight and exterior dimensions of selected car models marketed by U.S. and foreign manufacturers are given in Table 2-12, grouped by class, to indicate the status of vehicle design in 1978. These data represent the data base from which the vehicle characteristics in 1985-90 were projected. It should be noted that the class in which the vehicles are listed in Table 2-12 is not always the same as that designated by the manufacturer when the new vehicle design is introduced. For example, the Fairmont is termed a compact car by Ford, but in the context of the present study, it is clearly an excellent example of a mid-sized car of the 1980s.

Table 2-10. Summary of GM Transmission Study Computer Results

Number of Gears	Gear Speed-Ratio Range	Torque Converter Lockup	Increase in Optimum Fuel Economy ⁽¹⁾
MANUAL TRANSMISSIONS			
4 ⁽²⁾	3.56	N.A.	0 (baseline)
4 ⁽²⁾	4.72 (overdrive)	N.A.	3.8%
4	5.94	N.A.	4.4%
5 ⁽²⁾	4.72 (overdrive)	N.A.	4.7%
AUTOMATIC TRANSMISSIONS ⁽³⁾			
3 ⁽²⁾	2.84	No	0 (baseline)
3	2.84	Yes - 3rd gear	6.7%
4	4.22 (overdrive)	Yes - 3rd & 4th gear	14.3%
		Yes - all gears ⁽⁴⁾	18.0%
4	6.00	Yes - all gears ⁽⁴⁾	19.7%
(1) Composite fuel economy for optimum vehicle hp and axle ratio and shifting schedule; 0-60 mph acceleration in 13.5 seconds. (2) Stock transmission. (3) Torque converter stall torque ratio of 2-2.5. (4) Torque converter lockup at optimum efficiency condition in 1st gear. N.A. = Not applicable.			

Some of the new car designs utilize front-wheel drive and this trend is likely to continue in the years ahead as efficient packaging and weight saving become even more important. Measurements of the engine and the engine compartment in 1978 front-wheel drive vehicles are given in Table 2-13. The data have been used to estimate the volume box requirements sketched in Figure 2-13 for engines used in front-wheel drive vehicle designs.

2.4 PROJECTION OF VEHICLE CHARACTERISTICS AND ENGINE REQUIREMENTS FOR 1985

Before the baseline fuel economy of vehicles using conventional engines and transmissions can be projected, it is necessary to project the curb weight of the vehicles and the engine horsepower required for specified acceleration performance. Vehicle weights for the four classes defined previously were determined, starting with the design data

Table 2-11. Comparison of EPA Fuel Economy Results for Vehicles Using Manual and Automatic Transmissions

Inertia Weight, lb	Engine	Transmission	Axle Ratio	Fuel Economy, mpg ⁽¹⁾		
				City	Highway	Comp.
2250	L4	A3	3.76	23.5	34.5	27.4
	97 CID	M4	3.70	29	40	33.1
2500	L4	A3	3.91	25	33	28.1
	97 CID	M4	3.91	28	39	32.1
		M5	3.91	28	41	32.7
2750	L4	A3	3.18	23	32	26.3
	140 CID	M4	2.73	24.5	37	28.9
3500	V-6	A3	2.56	19	26	21.6
	231 CID	M4	2.56	17	27	20.4
		M5	2.56	19	31	23.0
3500	L6	A3	2.76	20	26	22.3
	225 CID	M3	3.21	20	28	23.0
		M4	3.23	20.5	31	24.2
3500 ⁽²⁾	LG	A3	2.71	18	26	20.9
	225 CID	A3, lockup	2.71	20	27	22.6
		M4 (OD)	3.21	20	32	24.1
4000 ⁽²⁾	V-8	A3	2.71	14	20	16.2
	318 CID	A3, lockup	2.45	15	22	17.5
		M4 (OD)	2.94	15	26	18.5
4500 ⁽²⁾	V-8	A3	2.71	14	20	16.2
	318 CID	A3, lockup	2.71	14	21	16.5

(1) 1977 certification data unless noted otherwise.

(2) 1978 certification data.

given in Table 2-12 and then applying weight reductions for each class based on engineering judgement. The results of this projection procedure are given in Table 2-14. The baseline drivetrain consists of a uniform-charge gasoline engine and a four-speed manual transmission or an automatic having equivalent overall gear speed ratio range and efficiency.

Table 2-12. Weights and Exterior Dimensions of
Downsized Passenger Cars

Manuf.	Model	Year Introduced ⁽¹⁾	Curb Weight, lb	Vehicle Dimensions, in.		
				L	W	H
VEHICLE CLASS: SMALL						
VW	Rabbit	1976	1860	155	63	55
Chevrolet	Chevette	1976	2050	162	62	52
Honda	Civic	1972	1762	150	59	52
Ford	Fiesta	1978	1775	147	62	52
Mazda	GLC	1977	1965	154	63	54
Toyota	Corolla		2055	165	62	55
Datsun	B-210		2020	162	61	54
Volvo	66	1977	1850	154	61	54
VEHICLE CLASS: COMPACT						
Audi	Fox		2100	174	65	54
VW	Dasher	1976	2200	173	63	54
Toyota	Corona		2535	173	64	54
Honda	Accord	1977	2018	163	64	52
Renault	12		2200	174	65	57
Volvo	343	1977	2154	166	65	55
Saab	99		2600	175	66	56
VEHICLE CLASS: MID-SIZED						
Ford	Fairmont	1978	2750	194	70	54
Chevrolet	Malibu	1978	3100	193	72	54
Ford	Granada	1975	3260	198	74	53
Dodge	Aspen	1976	3250	197	73	55
Audi	5000	1978	2725	190	70	54
Volvo	264		3170	193	67	56
Merc. Bz.	230		3200	191	70	56
VEHICLE CLASS: FULL-SIZED						
Chevrolet	Impala	1977	3700	212	76	56
Chrysler	Le Baron	1977	3525	204	74	53
Ford	LTD II	1977	4050	215	78	53

(1) The year of introduction is noted if the model represented a significant new design for the manufacturer rather than an evolution from previous designs.

Table 2-13. Engine and Engine Compartment Dimensions
for Downsized Vehicle Designs

Model	Vehicle Width	Engine/ Mount	Engine ^(1,2) Compartment L x W	Engine L x W ⁽¹⁾
VEHICLE CLASS: SMALL				
Fiesta	62	4 cylinders, transverse	24 x 30	17 x 13
Datsun F-10	60	4 cylinders, transverse	30 x 30	17 x 16
VEHICLE CLASS: COMPACT				
Audi Fox	65	4 cylinders, longitudinal	29 x 29	17 x 17
Saab 99	66.5	4 cylinders, longitudinal	30 x 33	20 x 20
VEHICLE CLASS: MID-SIZED				
Audi 5000	69	5 cylinders, longitudinal	30 x 30	22 x 19
Fairmont ⁽³⁾	70	6 cylinders, longitudinal	38 x 31	30 x 16
(1) All dimensions are given in inches. (2) Height is difficult to determine, but is between 24 and 28 in. (3) Rear-wheel drive; all other vehicles are front-wheel drive.				

The characteristics of the baseline 1985 vehicles are summarized in Table 2-15. The 0-60 mph acceleration time and power-to-weight ratio were selected to be consistent with the trend to reduced vehicle performance.

The vehicle characteristics projected represent a near-minimum size and weight for each class consistent with the specified passenger-carrying capacity and road performance. The extent to which such vehicle designs will actually appear on the market depends both on the public acceptance of downsized car designs and the need of the automobile industry for downsizing to meet mandated corporate average fuel

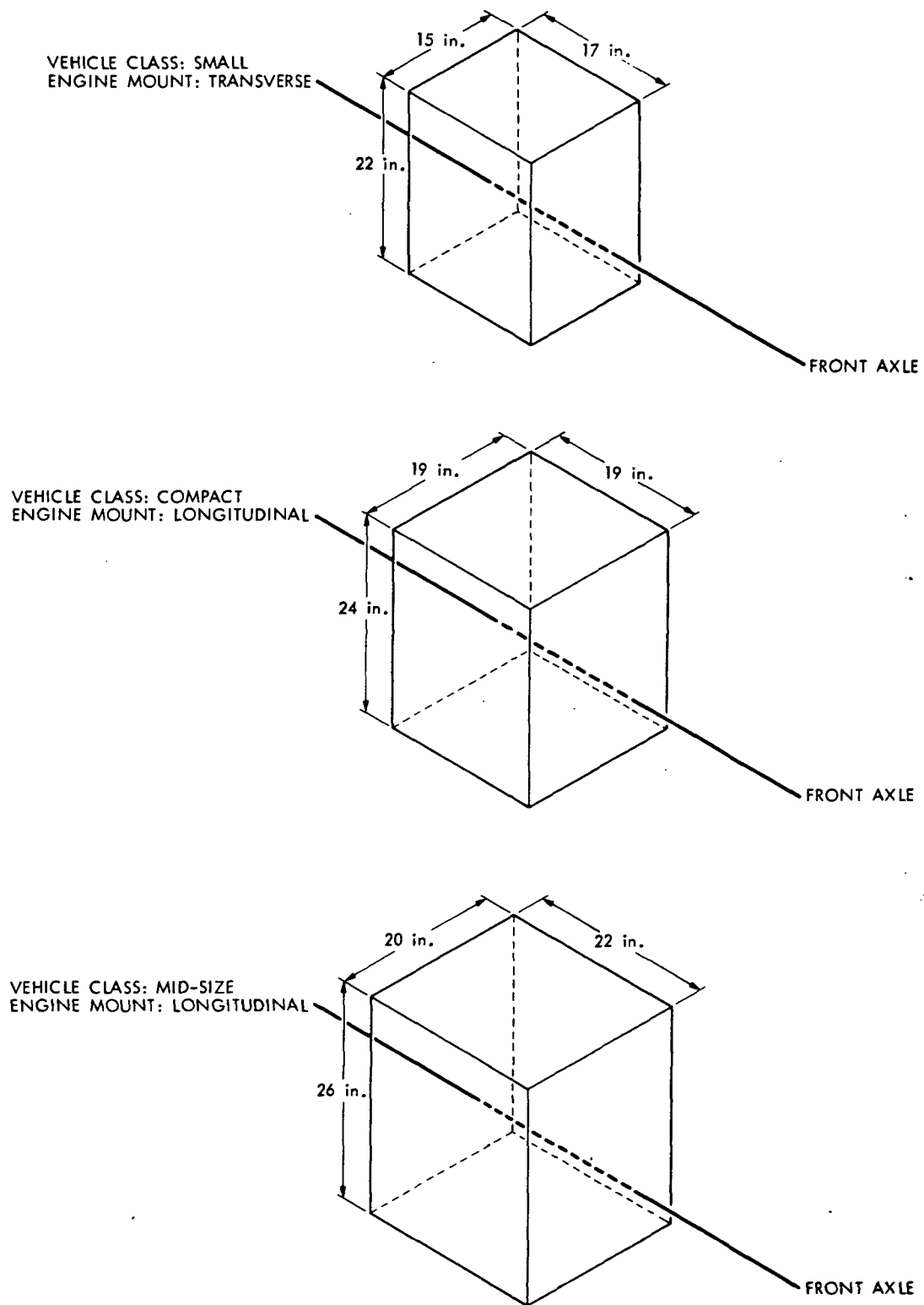


Figure 2-13. Volume Box Requirements for Engines
Used in Front-Wheel Drive
Vehicle Designs

Table 2-14. Vehicle Weights and Dimensions - 1978
and 1985 (Projected)

Year	Weight, lb		Dimensions, in.					
	Range	Average	Range			Average		
			L	W	H	L	W	H
VEHICLE CLASS: SMALL								
1978	1760-2055	1910	150-162	59-63	52-55	156	61	53
1985		1600				150	60	52
VEHICLE CLASS: COMPACT								
1978	2020-2600	2310	163-175	63-66	54-56	171	64	55
1985		2000				170	64	54
VEHICLE CLASS: MID-SIZED								
1978	2725-3260	3005	190-197	70-74	53-56	193	72	54
1985		2600				185	73	54
VEHICLE CLASS: FULL-SIZED								
1978	3700-400	3850	204-215	74-78	53-56	210	76	54
1985		3200				200	75	55

Table 2-15. Baseline Vehicle Characteristics for 1985 - Uniform-Charge
Gasoline Engine, Four-Speed Manual Transmission or
Equivalent Automatic

Vehicle Class	Length, in.	Curb Weight, lb	Inertia Weight, lb	0-60 mph Acceleration Time, s	Vehicle Power-to-Weight Ratio, hp/lb	hp
Small	150	1600	1850	18	0.024	44
Compact	170	2000	2300	16	0.026	60
Mid-Sized	185	2600	3000	14	0.029	87
Full-Sized	200	3200	3700	14	0.029	107

economy standards. As a result, a spectrum of vehicle sizes will be available in each class with the average weight, length, and horsepower in each class being somewhat greater than those given in Table 2-15.

2.5 PROJECTION OF BASELINE VEHICLE FUEL ECONOMY AND ENERGY INTENSITY FOR 1985

The baseline fuel economy for 1985 vehicles using conventional engines has been developed starting from the 1978 EPA fuel economy results (Reference 2-23) and data available from tests of prototype vehicles using advanced conventional engines such as turbocharged diesels (Reference 2-3). Fuel economy correlations as a function of vehicle inertia weight using 1978 engine and emission control technology are given in Figures 2-14 and 2-15 for the EPA urban and highway cycles. The significant effect of engine size on fuel economy for gasoline engines is clearly shown in the figures. In the context of this study, 1978 emission control technology means the use of three-way catalysts with and without EGR such that emission standards (0.4 g/mi HC, 3.4 g/mi CO, 1.5 g/mi NO_x) can be met with no fuel economy penalty relative to the 1978 Federal emission standards (1.5 g/mi HC, 15.0 g/mi CO, 2.0 g/mi NO_x). The 1978 baseline fuel economy for vehicles using naturally aspirated (NA) diesel engines is based on the EPA certification data for diesel engines given in Table 2-7. The fuel economy curve for vehicles equipped with turbocharged (TC) diesel engines is considerably less well established than for naturally aspirated (NA) diesels and is based on the prototype vehicle results of VW and Mercedes-Benz and the JPL computer simulations shown in Table 2-16. The fuel economy results shown in Figures 2-14 and 2-15 are for vehicles equipped with four-speed manual transmissions or automatics with equivalent overall efficiency.

It can be expected that gasoline engine systems will be further improved between 1978 and 1985 such that vehicles using gasoline engines will attain both improved fuel economy and reduced emissions during that time period. Present development activities on gasoline engines indicate that the engines of the 1985-90 period will utilize three-way catalyst, fuel injection, electronic control, and possibly increased compression ratio. In addition, it is possible that the advanced gasoline engine system will be augmented using either a turbocharger or supercharger so that a smaller displacement engine can be used without loss of the desired acceleration and passing performance. It is assumed in the present study that both the fuel economy and emissions of the boosted gasoline engines will be essentially the same as those of the naturally aspirated versions on the EPA city and highway cycles. Projected improvements in vehicle fuel economy resulting from the two gasoline engine developments expected between 1978 and 1985 are shown in Table 2-17 for vehicles with inertia weights between 2000 and 4000 lb. A modest fuel economy improvement of about 20% is projected between 1978 and 1985 for a fixed-inertia-weight vehicle. There is considerable uncertainty concerning the NO_x emission standard that such vehicles could meet without a fuel economy penalty. Based on the Volvo-Saab work using a three-way catalyst and fuel injection, it seems reasonable to expect that 2000-, 3000-, and 4000-lb (inertia weight) vehicles could meet a NO_x requirement of 0.4 g/mi, 0.6 g/mi, and 0.8 g/mi,

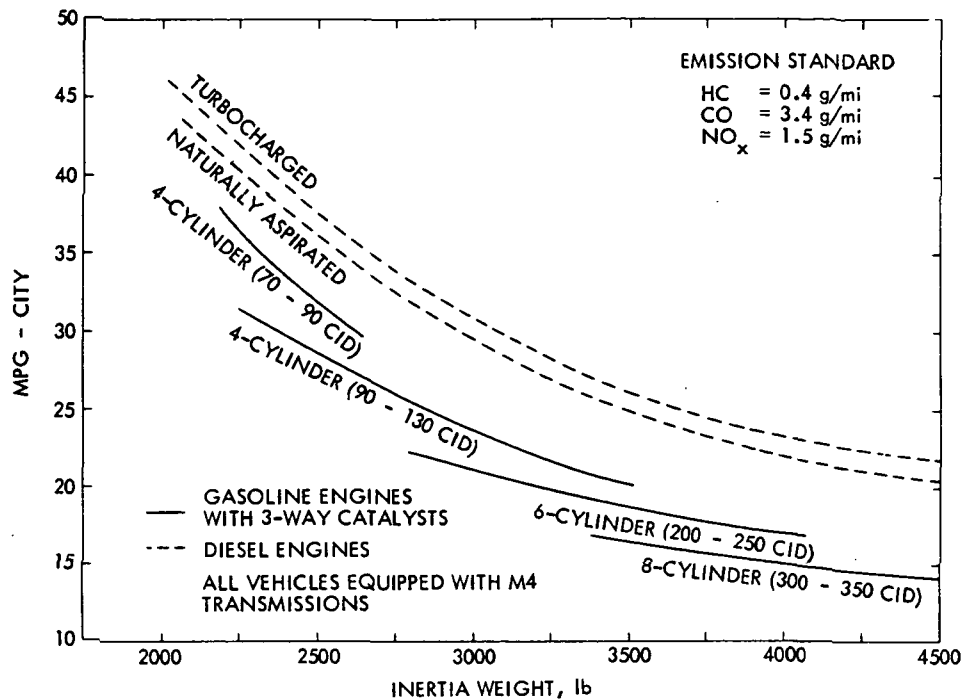


Figure 2-14. Baseline Fuel Economy - 1978 Technology, Urban Cycle

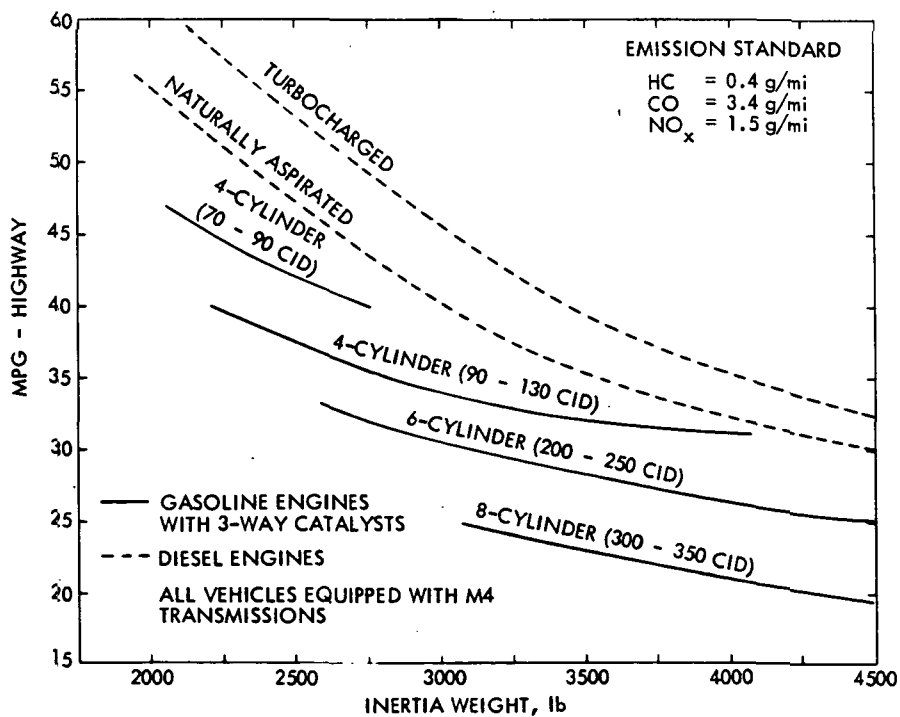


Figure 2-15. Baseline Fuel Economy - 1978 Technology, Highway Cycle

Table 2-16. JPL Computer Simulations of
Vehicle Fuel Economy

Vehicle Class	Inertia Weight, lb	Engine hp	Axle Ratio	Fuel Economy, mpg	
				Urban	Highway
ENGINE TYPE: (1) NA DIESEL (2)					
Compact	2000	50	3.9	42	54
Mid-sized	3000	65	3.9	33	41
Full-sized	4000	85	3.9	27	32
ENGINE TYPE: TC DIESEL (3)					
Compact	2000	60	3.5	42	55
Mid-sized	3000	95	3.5	31	37
Full-sized	4000	125	3.5	25	28
ENGINE TYPE: ROTARY (4)					
Compact	2000	60	3.5	30	42
Mid-sized	3000	95	3.5	23	31
Full-sized	4000	125	3.5	19	25
(1) All vehicles equipped with four-speed manual transmission (fourth gear is not overdrive).					
(2) NA diesel engine map as given in Reference 2-16.					
(3) TC diesel engine map as given in Reference 2-16.					
(4) Rotary engine map as given in Reference 2-6.					

Table 2-17. Projected Fuel Economy Improvements (%) in Internal
Combustion Gasoline Engines (1978-85)

Engine Development	Inertia Weight, lb		
	2000	3000	4000
Fuel injection, electronic control, and increased compression ratio	10	7	5
Boosted, smaller engine	10	15	15
Total projected improvement 1978-85	20	22	20

respectively. Further development of electronic control systems and three-way catalysts (efficiency and durability improvements) could result in lower NO_x emissions for the heavier vehicles.

Development of diesel engines for passenger cars is expected to continue with improvements in both vehicle fuel economy and emissions. All diesel engines presently used in production passenger cars are naturally aspirated, pre-chamber designs without emission control except possibly some injection timing retard. Current development activities indicate that turbocharged versions of several of the pre-chamber engines will be introduced within the next few years. In the case of the diesel engine, turbocharging improves both performance and fuel economy. The acceleration performance of turbocharged diesel-powered passenger cars would be comparable to that of many of the gasoline-fueled cars now being marketed. Additional diesel development activity (References 2-24 and 2-25) is directed toward a direct injection engine for passenger cars since such a combustion system would likely yield a 10-15% improvement in thermal efficiency over that of pre-chamber designs. Projected improvements in vehicle fuel economy resulting from the diesel engine developments cited are shown in Table 2-18. As for gasoline engines, a modest improvement of 15-20% between 1978 and 1985 is projected. Much work is currently being done to reduce NO_x emissions, odor, and smoke from diesel engines for passenger car use. Adaptation of EGR and electronic control are the major approaches being taken, but basic combustion system modifications are also being studied. Little has been published as yet concerning the results of the diesel emission reduction studies, but the VW vehicle results (Reference 2-3) using EGR with a turbocharged diesel indicate that meeting the same NO_x standards as suggested for the gasoline engines is possible for diesels. Whether or not those NO_x standards can be met without a fuel economy penalty or a smoke problem is highly uncertain at the present time.

The results obtained in this study for the moving fuel economy baseline for vehicles equipped with gasoline and diesel engines are given in Tables 2-19 and 2-20. The projected composite cycle fuel economy for light-duty vehicles in 1985 is shown in Figure 2-16 with the 27.5-mpg corporate average standard indicated for reference. The present results show that the 27.5-mpg standard can be met for inertia weights up to about 3500 lb using gasoline engines and up to about 4200 lb using diesel engines. It is also of interest to consider the projected energy intensity in 1985 of vehicles using gasoline and diesel engines. The results of the study in this regard are shown in Figure 2-17. The energy content of the gasoline and diesel fuels is noted on the figure along with the energy intensity equivalent (4350 Btu/veh-mi) of the 27.5-mpg gasoline standard. On an equivalent energy basis, the advantage of the diesel engine appears to be quite small at inertia weights of 2000-2500 lb, but it becomes significant (20-25%) for inertia weights of 3500-4000 lb.

Table 2-18. Projected Fuel Economy Improvements (%)
in Diesel Engines (1978-85)

Engine Development	Inertia Weight, lb		
	2000	3000	4000
Turbocharging			
City cycle	5	5	5
Highway cycle	10	10	10
Improvements in combustion in pre-chamber engines or introduction of direct injection engines	10	10	10
EGR and electronic control to reduce smoke and NO _x emissions	--	--	--
Total projected improvement 1978-85			
City cycle	15	15	15
Highway cycle	20	20	20

The 1985 baseline fuel economy and energy intensity for the four vehicle classes previously discussed (Table 2-15) are given in Table 2-21 for vehicles using turbocharged diesel and uniform-charge gasoline engines. The fuel economy values shown represent a baseline with which the fuel economy attainable in 1985-90 using the various alternate engines can be compared. In addition to the advanced alternative engines (gas turbine and Stirling), it is likely that, by 1985, significant progress will have been made on advanced conventional engines such as the PROCO stratified-charge engine, the spark-assisted, direct-injection diesel, and the direct-injected, stratified-charge rotary engine.

2.6 CONCLUSIONS

A review and assessment of engine, transmission, and vehicle design trends and EPA fuel economy and emissions data for 1973-78 indicate that by 1985 the size and weight of new vehicles in the four passenger car classes - small, compact, mid-sized, and full-sized - will be considerably reduced from most of those of today and that the fuel economy of the downsized vehicles using gasoline and diesel engines will be close to or exceed the 27.5-mph (composite cycle) standard for 1985. The engines required by the projected 1985 vehicles are compact, light, and of relatively low horsepower - less than 100 hp except for full-size vehicles, which would require 110-125 hp. Recent

Table 2-19. The Moving Fuel Economy Baseline in Vehicles
Equipped with Gasoline Engines

Vehicle Class:	Small	Mid-Sized	Full-Sized
Inertia Weight:	2000 lb	3000 lb	4000 lb
Transmission: (1)	M4	M4	M4
1973			
Engine Type	L4, 55 hp	L6, 105 hp	V8, 150 hp
City -----	26 mpg	17.5 mpg	12.4 mpg
Highway -----	--	--	--
Composite -----	--	--	--
1975 (2)			
Engine Type	L4, 55 hp	L6, 105 hp	V8, 150 hp
City -----	29 mpg	21 mpg	13.5 mpg
Highway -----	38	27	18.0
Composite -----	32.5	23.3	15.2
1978 (3)			
Engine Type	L4, 55 hp	L6, 105 hp	V8, 150 hp
City -----	40 mpg	23.3 mpg	15.0 mpg
Highway -----	50	31.5	21.5
Composite -----	44	26.4	17.4
1985 (4)			
Engine Type	L4, TC, 50 hp	L4, TC, 90 hp	V6, TC, 125 hp
City -----	46 mpg	28.5 mpg	18 mpg
Highway -----	57.5	38.5	26
Composite -----	50.5	32.3	20.9
NO _x -----	0.4 g/mi	0.6 g/mi	0.8 g/mi

- (1) All vehicles equipped with transmissions having overall torque ratios and efficiencies equivalent to M4 transmission.
- (2) Vehicles utilize an oxidation catalyst and EGR for emission control and meet Federal 1975 standards.
- (3) Vehicles utilize a carbureted fuel system and a three-way catalyst, O₂ sensor with feedback control, and EGR to meet the California 1978 standards.
- (4) Vehicles utilize a fuel-injection-type fuel system with electronic control, turbo- or supercharger boost, and a three-way catalyst and meet the Federal 1985 standards except as noted for NO_x.

Table 2-20. The Moving Fuel Economy Baseline in Vehicles Equipped with Diesel Engines

Vehicle Class:	Small	Mid-Sized	Full-Sized
Inertia Weight:	2000 lb	3000 lb	4000 lb
Transmission:(1)	M4	M4	M4

1978 ⁽²⁾			
Engine Type	L4, 50 hp	L5, 80 hp	L6, 100 hp
City -----	45 mpg	30 mpg	22 mpg
Highway -----	55	40	32.5
Composite -----	49	33.8	25.7

1985 ⁽³⁾			
Engine Type	L4, TC, 50 hp	L5, TC, 90 hp	V6, TC, 125 hp
City -----	52 mpg	34.5 mpg	25.3 mpg
Highway -----	66	48	39
Composite -----	57.5	39.5	30
NO _x -----	0.4 g/mi	0.6 g/mi	0.8 g/mi

- (1) All vehicles equipped with transmissions having overall torque ratios and efficiencies equivalent to M4 transmission.
- (2) Vehicles utilize no exhaust treatment and meet Federal 1978 standards.
- (3) Vehicles utilize turbocharged engines with EGR and electronic control to control smoke, odor, and NO_x emissions and meet the Federal 1985 standards except as noted for NO_x.

developments in the packaging of both gasoline and diesel engines, such as increasing engine rpm, material substitution to reduce weight, and turbocharging, have significantly changed the baseline of the conventional engines and resulted in engines quite compatible with the down-sized vehicle designs.

In the areas of fuel economy and emissions, there have also been significant improvements in the current(1978) and projected (1985) capability of the conventional engines. Composite fuel economies of 54, 43, 34, and 26 mpg using gasoline engines, and 60, 47, 38, and 32 mpg using diesel engines have been projected for small, compact, mid-sized, and full-sized cars in 1985, respectively. Those fuel economies

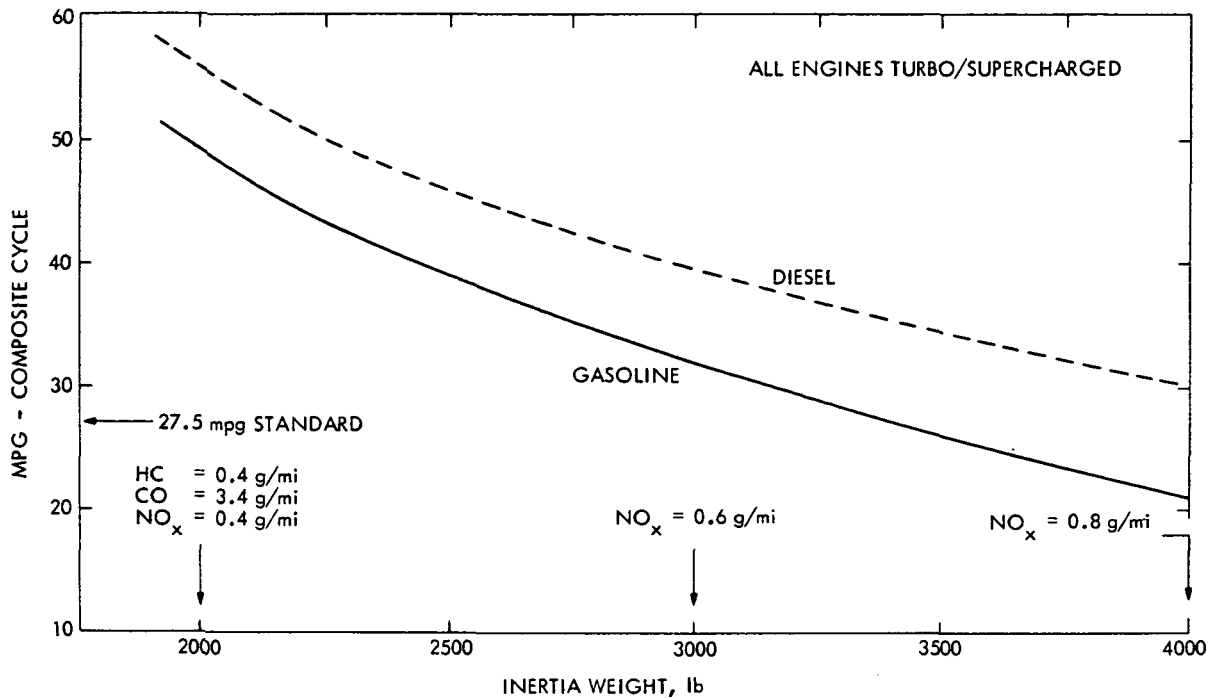


Figure 2-16. Projected Composite Fuel Economy for Light Duty Vehicles in 1985 Using Gasoline and Diesel Engines

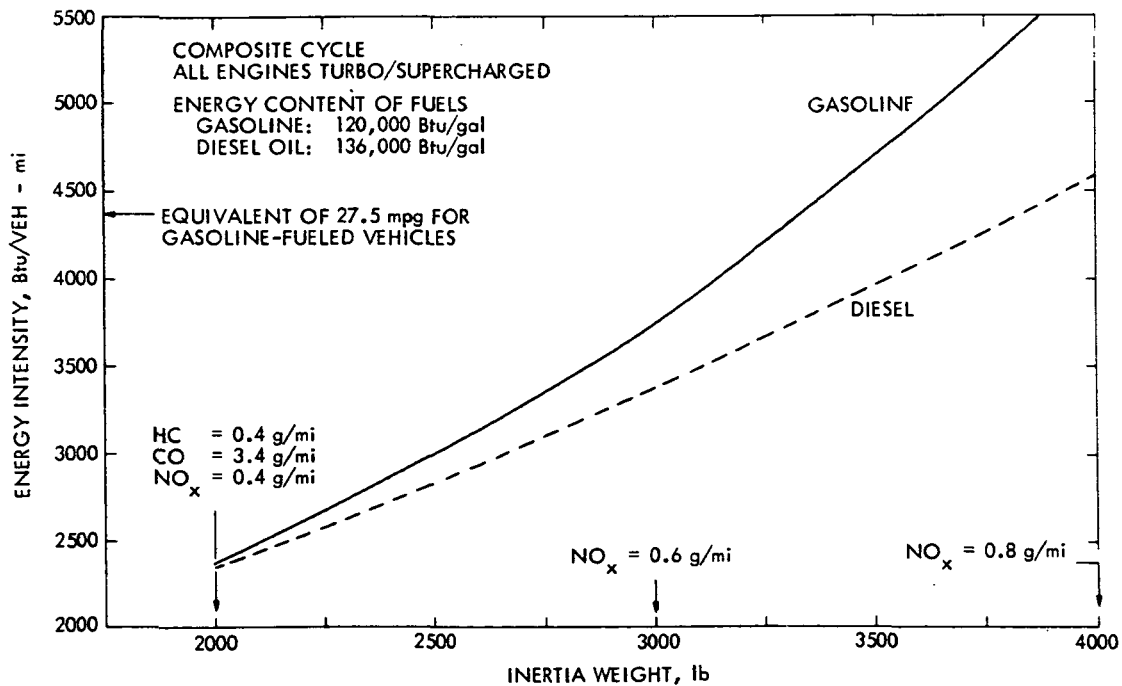


Figure 2-17. Projected Energy Intensity of Light Duty Vehicles in 1985 Using Gasoline and Diesel Engines

Table 2-21. 1985 Baseline Fuel Economy and Energy Intensity Using Uniform-Charge Gasoline and Diesel Engines

	Gasoline	TC Diesel
VEHICLE CLASS: SMALL		
Engine Type:	4/44/M4 ⁽¹⁾	4/50/M4 ⁽¹⁾
Inertia Weight:	1850 lb	1950 lb
City -----	49	54
Highway -----	61	68
Composite -----	53.8	59.5
Btu/veh-mi -----	2230	2285
VEHICLE CLASS: COMPACT		
Engine Type:	4/60/M4 ⁽¹⁾	4/68/M4 ⁽¹⁾
Inertia Weight:	2300 lb	2450 lb
City -----	39.5	42
Highway -----	50	55.5
Composite -----	43	47.2
Btu/veh-mi -----	2791	2881
VEHICLE CLASS: MID-SIZED		
Engine Type:	6/87/M4 ⁽¹⁾	5/98/M4 ⁽¹⁾
Inertia Weight:	2900 lb	3090 lb
City -----	30	33
Highway -----	40	47
Composite -----	33.8	38.1
Btu/veh-mi -----	3550	3570
VEHICLE CLASS: FULL-SIZED		
Engine Type:	6/107/M4 ⁽¹⁾	6/122/M4 ⁽¹⁾
Inertia Weight:	3500 lb	3730 lb
City -----	22.2	27
Highway -----	32	41
Composite -----	25.7	31.9
Btu/veh-mi -----	4669	4263
(1) Cylinder/hp/transmission.		

are about 20% greater than those being achieved (EPA certification data) in 1978 by four-speed manual-transmission-equipped vehicles of the same inertia weight and engine displacement. With the development of the three-way catalyst, it appears that NO_x emissions of less than 1.0 g/mi (0.8, 0.6, 0.4 g/mi for full-, mid-, compact-, and small-sized cars, respectively) can be achieved using turbocharged gasoline engines. Whether these same low NO_x emissions can be met using turbocharged diesel engines is unclear at the present time (1978), but initial results with small diesel engines are encouraging.

SECTION 3

GAS TURBINES

3.1 INTRODUCTION

Theoretically, the Brayton cycle represents one of the most efficient and practicable methods of converting heat into mechanical power. Particularly attractive qualities of engines implementing a Brayton cycle are their potential for low pollution and for utilizing a variety of fuels. In a Brayton cycle, a gas is compressed and heated at constant pressure, then expanded to the initial pressure of the cycle. The original Brayton engine (1873) utilized separate cylinders for compression and expansion and a valve-controlled combustion chamber. The Brayton engine of interest for automotive power is atmospherically-aspirated utilizing an aerodynamic compressor and turbine with heat introduced into the flow by means of combustion. In this form the Brayton engine is commonly referred to as the "gas turbine engine".

As a result of technological advances in the aviation field during and after World War II, gas turbines found their way into the transportation field wherever the power profile was fairly constant and predictable, and where engine specific weight and size was of importance. Gas turbines have now entirely replaced piston engines on transport aircraft and helicopters and are also used successfully to provide propulsive power for boats, trains, and hovercraft. The development of gas turbines for automotive vehicles has been progressing at a much slower pace. Utilizing, at first, spin-off technology from the aviation field, a number of car and engine manufacturers embarked on the development of automotive gas turbines in the early 1950s. Considerable progress has been made. Components, materials, and production technology have been improved and progress has been made in understanding vehicle-gas turbine interactions. A large variety of automotive gas turbines have been built and tested during the last three decades in trucks, buses, and cars. In one case (Chrysler) 50 turbine-powered cars were given to consumers on an experimental trial basis in order to obtain broad-based statistical data. None of these efforts, however, has resulted in a mature, viable, and marketable product capable of competing with the conventional piston engine.

Gas turbines have had difficulty competing in the automotive field because of a number of special problems that arise from the operating conditions and fuel economy requirements of automobiles. Road vehicles, in particular passenger cars, operate at a fraction of their design power most of the time, and at such operating conditions gas turbines tend to be less efficient than comparable piston engines. In automobile engine design, cost-effectiveness considerations dictate simple design approaches; the degree of complexity acceptable for aircraft is not economically viable. New components such as an efficient regenerative heat exchanger and special engine control systems were not available from the other applications and had to be developed from scratch. A problem that strongly affects the rate of development of vehicular gas turbines is that of thermal stress and fatigue. These

are caused by the frequent and rapid variations of operating temperatures, which are unavoidable in an automotive application, but which represent deviations from the optimum design operating conditions of the turbine and the compressor. Inertial losses occur whenever the rotary masses have to be accelerated. The end result is that the average fuel economy that can be obtained with gas turbines under actual driving conditions is significantly impaired. A problem that particularly retards the development of small gas turbines for passenger cars is that of engine size. Below 200-300 hp the fuel and weight efficiency of gas turbines begins to decline and may drop even more rapidly below 100 hp. Most automotive turbine development effort has therefore been directed toward applications in trucks and buses instead of passenger cars. During recent years the problem of meeting emission standards has been added, complicating the overall problem.

The worldwide interest in energy conservation and clean environments has drawn renewed government and industrial attention to the automotive gas turbine with particular reference to its potential for use in passenger cars. It has been recognized that there is a need for more fundamental research and for concerted, well-coordinated development efforts, instead of the isolated efforts made in the past. Private and government-funded research and development is in progress in the United States, Europe, and Japan. These efforts are particularly concerned with the needs for higher cycle temperature, improved part-load fuel economy, and cost-effective production.

3.2 ENGINE CONCEPTS

3.2.1 Basic Thermal Concepts

The basic design parameters of the gas turbine engine are the turbine inlet temperature and the compression ratio. These parameters, in combination with the degree of regeneration, determine the specific fuel consumption and the power output that can be obtained per unit of mass flow through the engine. Figure 3-1 shows the general relationships between these parameters for state-of-the-art component efficiencies. The breaks in the lines of constant turbine temperature represent the points beyond which (assuming moderately loaded radial compressors) two-stage compression is needed, which is at a compression ratio of about 5. Plotted on the right is the specific fuel consumption obtainable with state-of-the-art diesel and spark-ignition engines. It can be seen that turbine engine performance and fuel economy increase with turbine inlet temperature regardless of whether heat is recovered or not. Heat recovery (regeneration), however, has a very strong effect on fuel economy at all turbine temperatures. It is important to note that the compression ratios required to obtain optimum fuel economy for a regenerative engine are low (about 4), and are well within range of a single-stage radial compressor. Based upon these general relationships, it was concluded early (in 1950) that the most reasonable approach to satisfying the conflicting demands for competitive fuel economy, mechanical simplicity, reliability, and cost would be the engine thermal concept shown schematically in Figure 3-2: turbine inlet

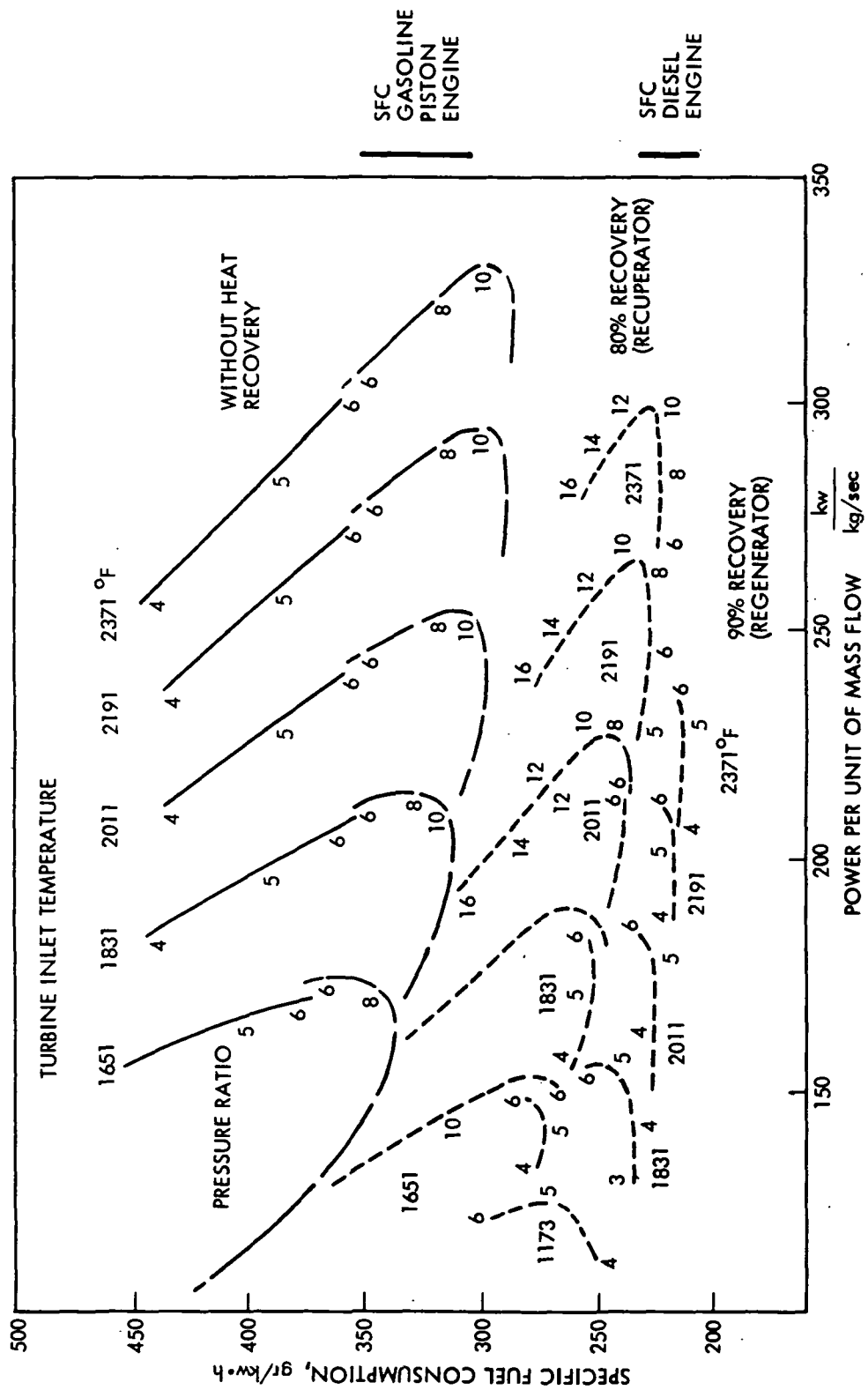


Figure 3-1. Relationship Between Primary Design Criteria for Gas Turbines (Reference 3-1)

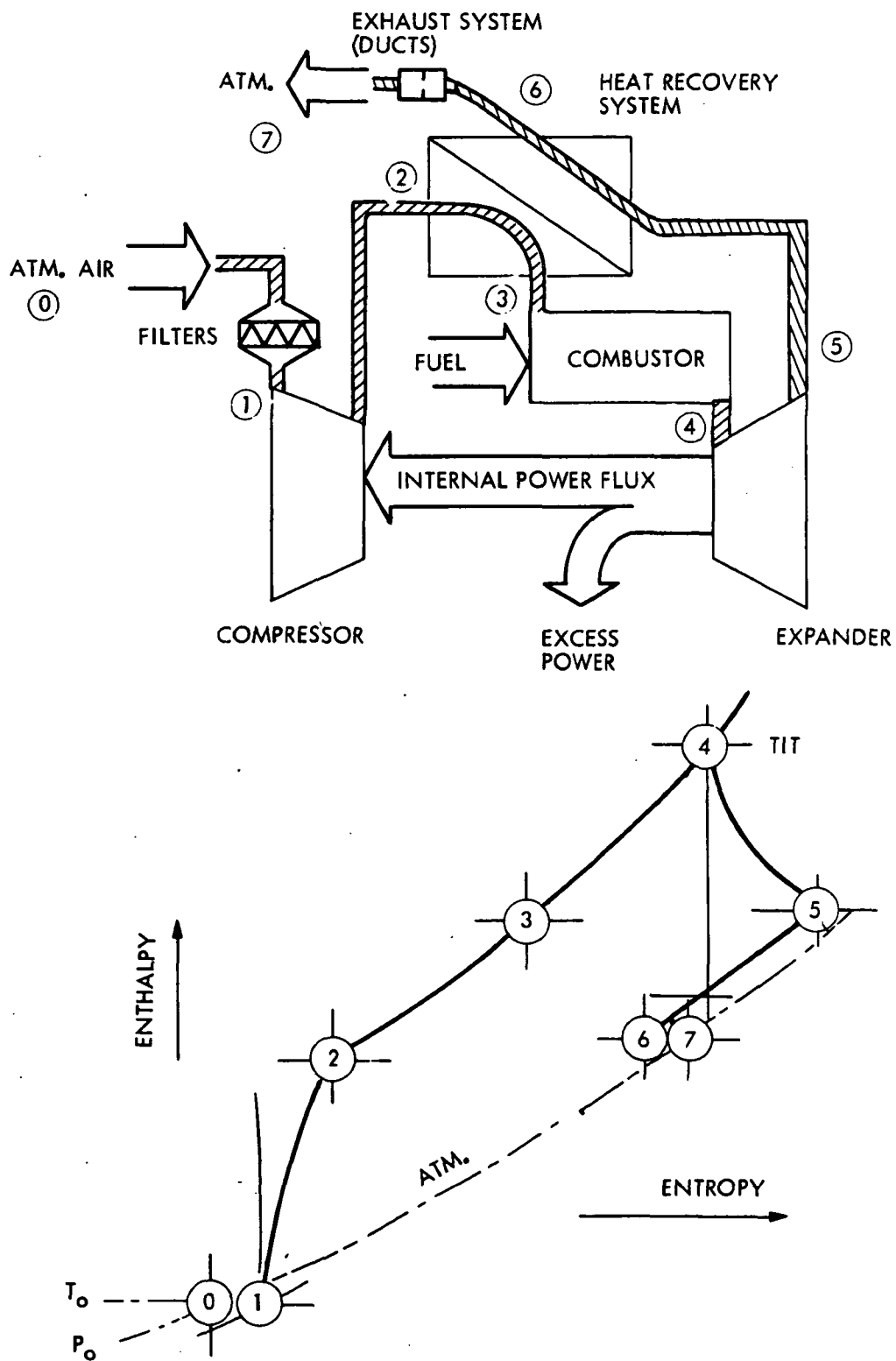


Figure 3-2. Schematic of Thermal Concept of Typical Automotive Gas Turbines

temperature as high as creep-rupture and service-life considerations would allow, a low compression ratio producible by one radial compressor stage, and regenerative heat recovery from the exhaust.

There are two approaches to the implementation of the heat recovery: rotary (regenerators) and static heat exchangers (recuperators). Rotary (disc type) heat exchangers are to date the preferred choice of most developers. Developments with recuperative heat exchangers are also under way in order to circumvent the leakage and seal life problems that exist with rotary regenerators.

3.2.2 Gas Turbine Configurations

In the course of efforts to improve drive cycle fuel economy and engine response, a variety of approaches have been pursued with regard to the arrangement of major components, shafting, and power take-off. These will be discussed in the following paragraphs.

3.2.2.1 Two-Shaft (Free-Turbine) Engines. The two-shaft engine (Figure 3-3) has been the preferred configuration of most developers during the past three decades. It consists of an axial turbine that drives a radial compressor (referred to also as the "gasifier"), and a separate single or multistage axial turbine that delivers power to the drive train from the turbine exhaust side. Two-shaft engines with a radial inflow turbine followed by an axial free power turbine have also been developed (Reference 3-2, p. 259). Such an arrangement, however, results in a long engine because of the diverging transition duct required between the turbines. The major advantage of the two-shaft (or free-turbine) engine over a single-shaft engine is the torque-speed characteristic which, as is shown in Figure 3-4, is more favorable for automotive operation. The torque range attainable with the power turbine at a constant gasifier speed is about 2, which makes it possible to cover the drive range of passenger cars with at most three external gears. The total expansion required from a compression ratio optimum for a regenerative cycle (about 4) can be handled by two axial flow turbines (gasifier and power) having aerodynamic and circumferential speed characteristics acceptable from the standpoint of stress and efficiency.

Disadvantages of the free-turbine system encountered during early road tests of such engines were inadequate braking and the possibility that the power turbine could overspeed (i.e., "run away") in situations where there is a loss of traction (wheel spin) or drive train malfunctions. This deficiency was eliminated by the introduction of the cross-drive (Figure 3-3), which connects the power turbine with the gasifier through an overrunning clutch. This allows the compressor to function as a power attenuator for braking. A braking capability far exceeding that of piston engines can be achieved with this arrangement, particularly if compressor air is bled off to control braking. The cross-drive feature was introduced in the early 1950s by Rover (Reference 3-2) and was adopted by other developers who were active at that time.

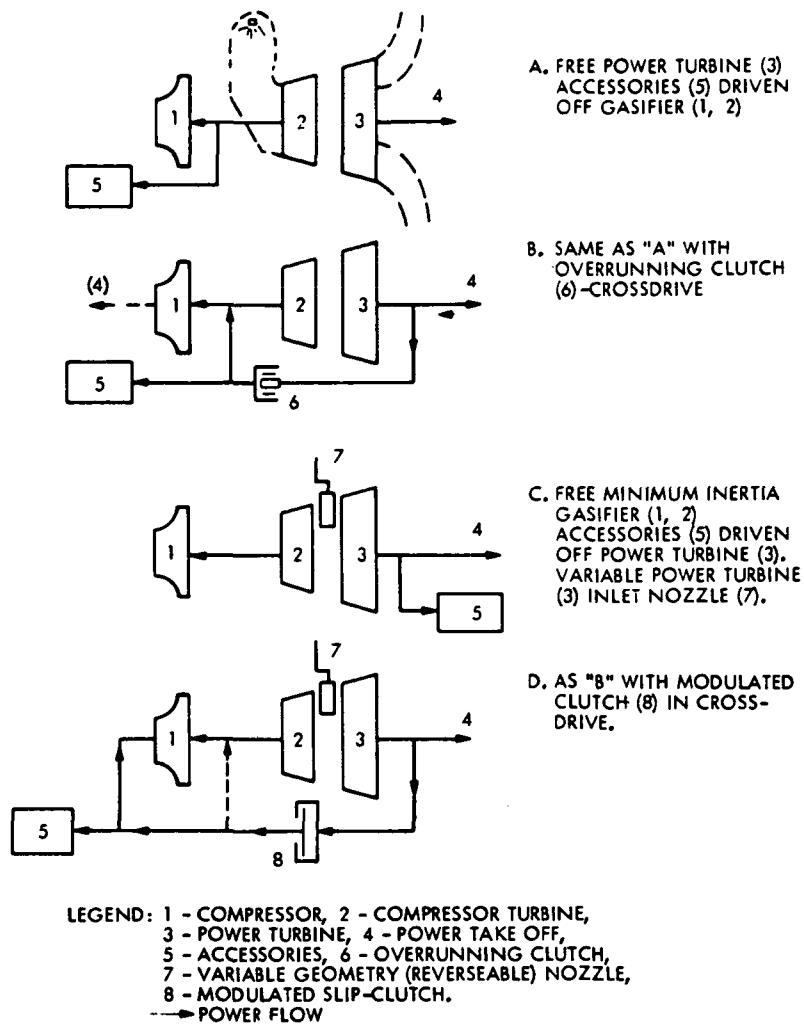
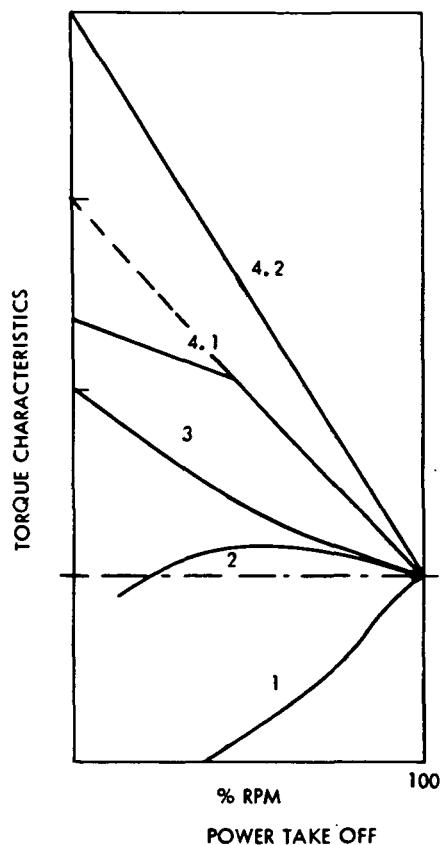


Figure 3-3. Two-Shaft Configurations for Automotive Gas Turbines

Another approach to achieve cross-drive was taken by Volkswagen (Reference 3-2, p. 448). It utilizes a concentric shaft arrangement between the power turbine and the gasifier with power take-off from the compressor side. This eliminates the need for a long extra driveshaft between the power turbine and the compressor and has the additional advantage of unrestricted space for the design of an efficient turbine exhaust diffuser. The disadvantages of this design approach, however, are the need for relatively large gasifier bearings, higher parasitic losses, and a higher gasifier inertia. This approach does lead to a convenient rear engine installation as in the VW Fastback and Microbus (Figure 3-5).



- 1 - SINGLE SHAFT ENGINES
- 2 - DIESEL
- 3 - TWO SHAFT FREE TURBINE ENGINES
- 4 - TURBINE INTEGRATED (3 - SHAFT) TRANSMISSION SYSTEMS
- 4.1. FIAT, BMW, --- FLY WHEEL AUGMENTED (REF 3-2)
- 4.2. KRONOGARD KTT. (REF 3-3)

Figure 3-4. Comparison of Torque Characteristics for Various Gas Turbine Engine Configurations

Other problems that became apparent during the road testing of automotive gas turbines were insufficient gasifier response, overheating of the regenerator heat exchanger during engine acceleration, and internal losses caused by off-design operating conditions during rapid changes in engine speed and power. These off-design excursions were thought to be primarily responsible for the discrepancies found between fuel economy results predicted using engine dynamometer data and those experienced in actual driving. Attempts to alleviate this problem led to the use of variable guide-vanes at the power turbine inlet. They were first introduced by Chrysler on their second generation engine in 1955. The rotation of the guide-vanes causes a change in swirl and the nozzle throat area which controls the back pressure of the gasifier turbine as well as its speed and performance. This results in a shift of power to the gasifier turbine for improved acceleration of the gasifier rotor and is necessary to keep the cycle

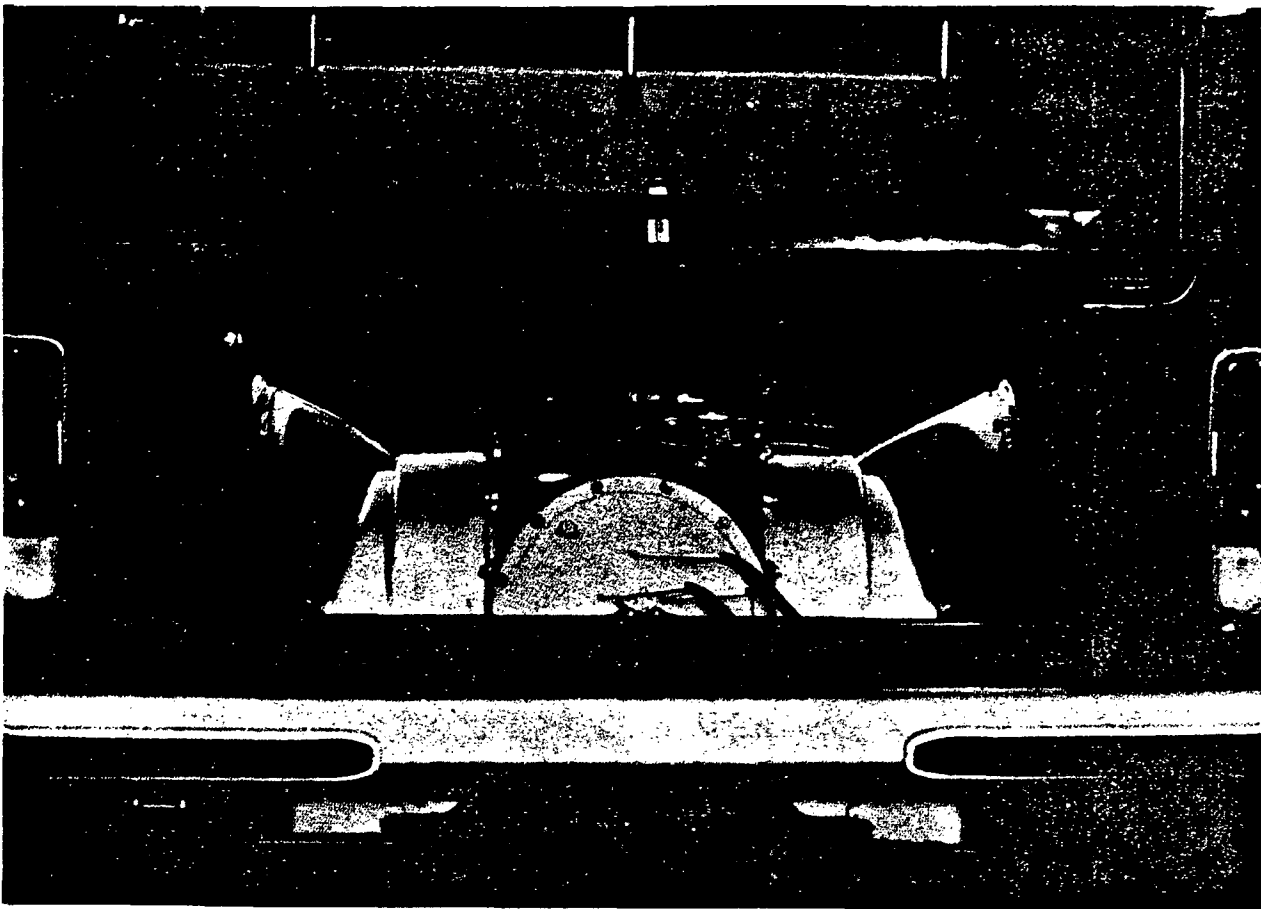


Figure 3-5. VW GT-70 in Microbus (Reference 3-4)

temperature up during part-load and idling conditions without exceeding the safe operating temperature of the regenerator.

To further improve engine response, Chrysler (Reference 3-5) went to a "free gasifier" engine (Figure 3-3C) to reduce gasifier rotor inertia to a minimum. This was accomplished by making the power turbine inlet guide-vanes reversible for braking and eliminating the cross-drive between the power turbine and the gasifier. The accessories were then driven off of the power turbine instead of the gasifier. With this arrangement, the power turbine had to be disconnected from the drive train to permit idling during vehicle stops. To avoid the major design effort that would have been necessary to incorporate a slip-clutch of sufficient capacity into the system, Chrysler used a three-speed gearbox and a hydrodynamical torque converter. Chrysler used this transmission approach in both their sixth and seventh generation engines (Reference 3-5). Other developers of free-turbine engines have preferred to use the cross-shaft and to drive the accessories off of the gasifier in order to utilize the inherent torque conversion capability of a free power turbine.

One of the latest innovations in shafting concepts is the "power transfer system" introduced by General Motors (Reference 3-6). As is shown in Figure 3-3D, it consists of a modulated slip-clutch in the cross-drive between the gasifier and the power turbine. The clutch pressure is regulated by means of a control device that senses combined clutch and accessory drive torque as well as shaft speed. Thus, this provides the optimum power distribution between the gasifier and the power turbine. The clutch also controls the gasifier to prevent the power turbine from overspeeding and protects the regenerator from overheating during power-braking with the power turbine nozzles in the braking position.

3.2.2.2 Single-Shaft Engines. Single-shaft engines are attractive because of their apparent mechanical simplicity and cost-effectiveness. Only a few single-shaft engines, however, have been developed, most of them with objectives other than automotive applications in mind. A major obstacle to the use of the single-shaft configuration is its inherent speed-torque characteristic (Figure 3-4), which requires the use of a continuously variable transmission (or a large number of gear ratios) in order to cover the torque range of an automobile. The present status of CVT development is discussed in Section 5.

Since the radial compressor is the preferred choice for automotive gas turbines, the primary variation in the design of a single-shaft engine lies in the choice of the turbine and combustor arrangement. Prospective arrangements under consideration are shown schematically in Figure 3-6. Arrangement A (top) essentially adopts the conventional two-shaft engine design except that the power turbine is on a common shaft with the compressor and power takeoff is from the front or the side instead of from the rear through the exhaust diffuser. The back-to-back arrangement (Concept B) of a radial compressor and a radial turbine is the concept envisioned by most engine designers because of its simplicity and the favorable torque characteristics and potential for wide-range variable-geometry control of radial turbines. Arrangement C utilizes a reversed axial turbine in line with a barrel combustor and a centerbody inlet. The advantages claimed for this concept are a minimum of total surface exposed to heat and a relatively small rotor volume and inertia when compared to a radial turbine with a scroll or a plenum inlet. From the failure probabilistic (Weibull) standpoint, this concept has the best potential for the introduction of ceramics and thus high turbine inlet temperature.

Despite the fact that a suitable CVT transmission was not available, the single-shaft concept was investigated in depth by General Electric, AiResearch, and United Aircraft Research Laboratories (References 3-7 through 3-9). About 1972, General Motors (DDA) started the development of an experimental single-shaft engine (designated AGT-2). This has become a pilot project for wide-range variable-geometry control; the objective is to demonstrate the feasibility and potential of constant-speed power control.⁽¹⁾ Because of its above-mentioned advantages for

(1) DOE has recently initiated a program with DDA to test the AGT-2 engine.

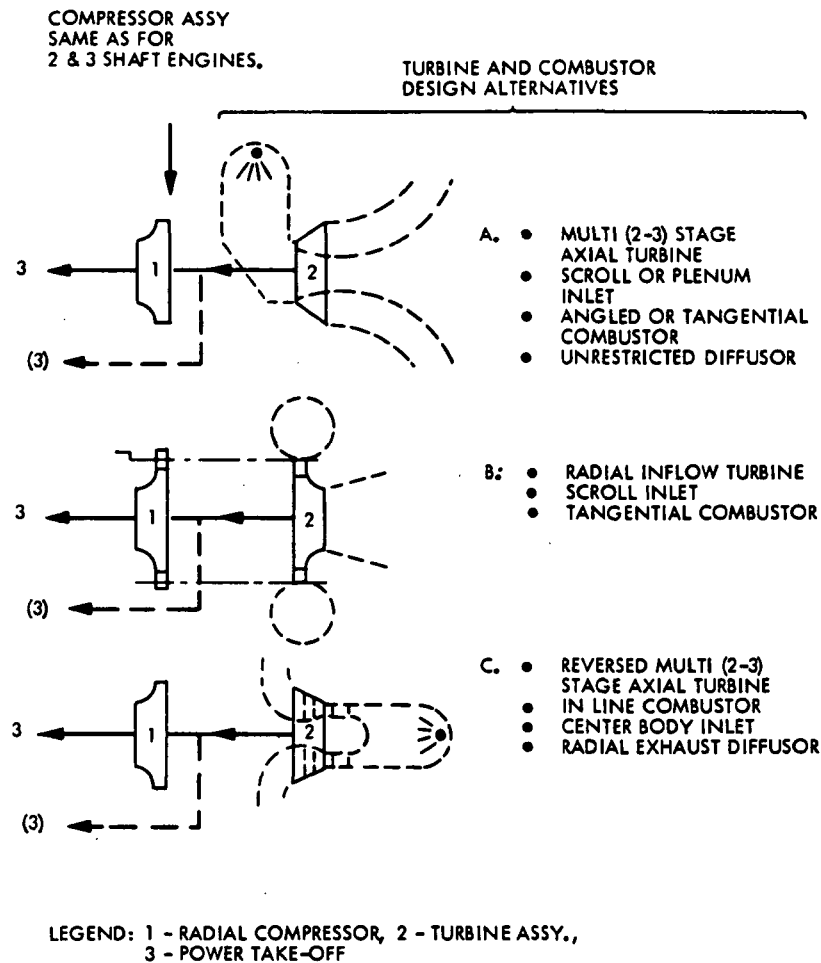
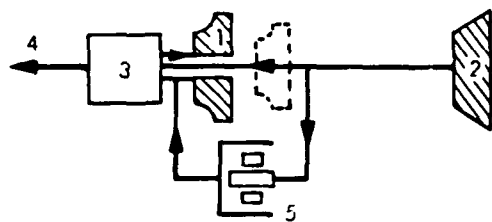


Figure 3-6. Prospective Single-Shaft Concepts for Automotive Gas Turbines

ceramics, the Ford Motor Company is developing a ceramic reversed axial turbine (Arrangement C) to demonstrate the feasibility of a high-temperature all-ceramic turbine (Reference 3-10).

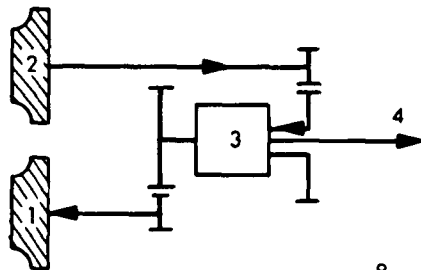
3.2.2.3 Multishaft Turbine Transmission Systems. Turbine transmission systems are multishaft (mostly three-shaft) systems where the inherent speed-torque characteristics of aerodynamic compressors and turbines are combined with those of differential gears. A variety of geared turbine systems have been conceived with different objectives in mind. Schematically shown in Figure 3-7 are typical systems that are currently under consideration for automotive application.

In Concept A, a single-shaft radial or an undivided multi-stage axial turbine drives the compressor (or a part of it) and the output shaft through a differential gear. The objective is to obtain a torque speed characteristic that otherwise could only be obtained with

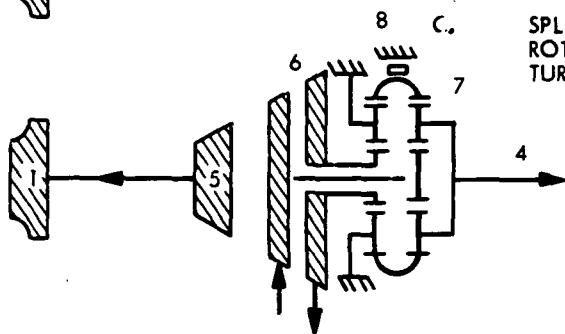


A. CONCENTRIC SHAFT
ARRANGEMENT
DIFFERENTIAL GEAR
AND OVERRUNNING
CLUTCH CROSS-DRIVE
(BMW)

- 1 - COMPRESSOR
- 2 - TURBINE
- 3 - DIFFERENTIAL GEAR
- 4 - POWER OUTPUT
- 5 - CROSS-DRIVE

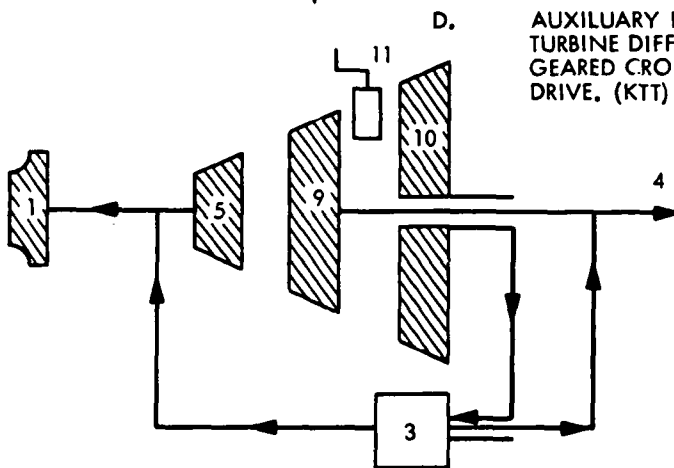


B. PARALLEL SHAFT
ARRANGEMENT.
DIFF. GEAR CROSS
DRIVE (FIAT)



C. SPLIT COUNTER-
ROTATING POWER
TURBINE (VOLVO)

- 5 - GASIFIER TURBINE
- 6 - SPLIT POWER TURBINE
- 7 - DIFFERENTIAL GEARS
- 8 - OVERRUNNING CLUTCH



D. AUXILIARY POWER
TURBINE DIFF.
GEARED CROSS-
DRIVE. (KTT)

- 3 - DIFFERENTIAL GEAR
- 9 - POWER TURBINE
- 10 - AUXILIARY TURBINE
- 11 - VARIABLE TURBINE NOZZLES

Figure 3-7. Turbine Transmission Systems

a free power turbine. Torque conversion takes place within the system owing to a power shift between the compressor (1) and the output shaft (4), which can come to a full stop. A cross-drive (5) between the compressor and the turbine is provided to prevent runaway of the turbine. The engine operates like a single-shaft engine whenever the cross-drive is locked. The torque characteristic of this concept is shown schematically in Figure 3-4. It has been successfully applied in small gas turbine starters for large turboprop engines. An automotive turbine engine projected by the Bavarian Motor Works (BMW) in the mid-1950s was also based on the concept.

Functionally, Concept B is the same as Concept A, the only difference being that a parallel instead of concentric shafting arrangement is used. The concept was originated by Fiat in the mid-1950s and was incorporated into the design of the Fiat TDC 804 2 gas turbine (Reference 3-2, p. 480) which was built and tested during the 1960s.

Concept C represents essentially a two-shaft engine that consists of a gasifier and a split power turbine. The power turbine is divided into two counterrotating turbines that are connected by planetary gears with an overrunning clutch between them. At speeds above 50 percent of the nominal speed, the second stage of the power turbine windmills in the exhaust flow of the preceding stage. Below 50 percent of the nominal speed, the second stage locks and extracts power from the exhaust swirl of the preceding stage. It is claimed in Reference 3-11 that a torque ratio of about 6:1 can be obtained with this arrangement. A disadvantage is the drag of the second stage during idling, and the complexity of the concentric shaft bearing arrangement.

Configuration D shows a concept referred to in recent publications (References 3-3, 3-12, 3-13, 3-14) as the KTT (Kronogard Turbine Transmission) system. The KTT engine has a third auxiliary turbine (10) added behind the power turbine, which connects to the power turbine and to the gasifier in a fixed torque ratio through a differential gear (3). The expansion ratio of the auxiliary turbine can be controlled by means of a variable nozzle (11). The basic advantages claimed for this approach are that the torque conversion capability of a two-shaft engine can be retained while simultaneously solving the gasifier inertia and accessory drive problem. The power turbine (9) can come to a full stop and does not have to be disconnected for idling as is necessary with a two-shaft free gasifier engine. Compared to a two-shaft engine, internal loss recovery is improved because a part of the torque conversion takes place downstream in the auxiliary turbine, which is always in motion. The developer (United Turbine AB, Sweden) also claims that the concept makes it possible to better optimize turbine aerodynamics; reduce turbine stress levels, cost, and size; keep cycle temperature and pressure up (longer) during part-load operations; and produce a torque ratio capability of about 4, requiring only a two-speed external gearbox. A disadvantage of the KTT system, however, is the complexity of the bearing arrangement and the need for three separate turbine rotors. This may significantly impair the potential of the concept for the introduction of ceramics.

The operation and efficiency of a gas turbine engine (Figure 3-8) depends largely upon the efficiency and proper matching of its major components. Data for typical state-of-the-art non-size-critical gas turbine components and sensitivities, in terms of the change in efficiency or operating condition required to produce a 1 percent improvement in the engine's specific fuel consumption, are summarized in Table 3-1. It can be seen from the table that engine efficiency is very sensitive to turbine inlet temperature and to leakage through the regenerator cross-seals or to air bypassing the turbine nozzle for the cooling of turbine discs and bearings. Pressure losses caused by stationary components such as transition housings and ducts also have a strong effect on engine efficiency, indicating a definite need for a high degree of aerodynamic quality throughout the engine. The compressor is the most sensitive major component, followed by the compressor turbine, the regenerator heat exchanger, the combustor, and (where applicable) the power turbine.

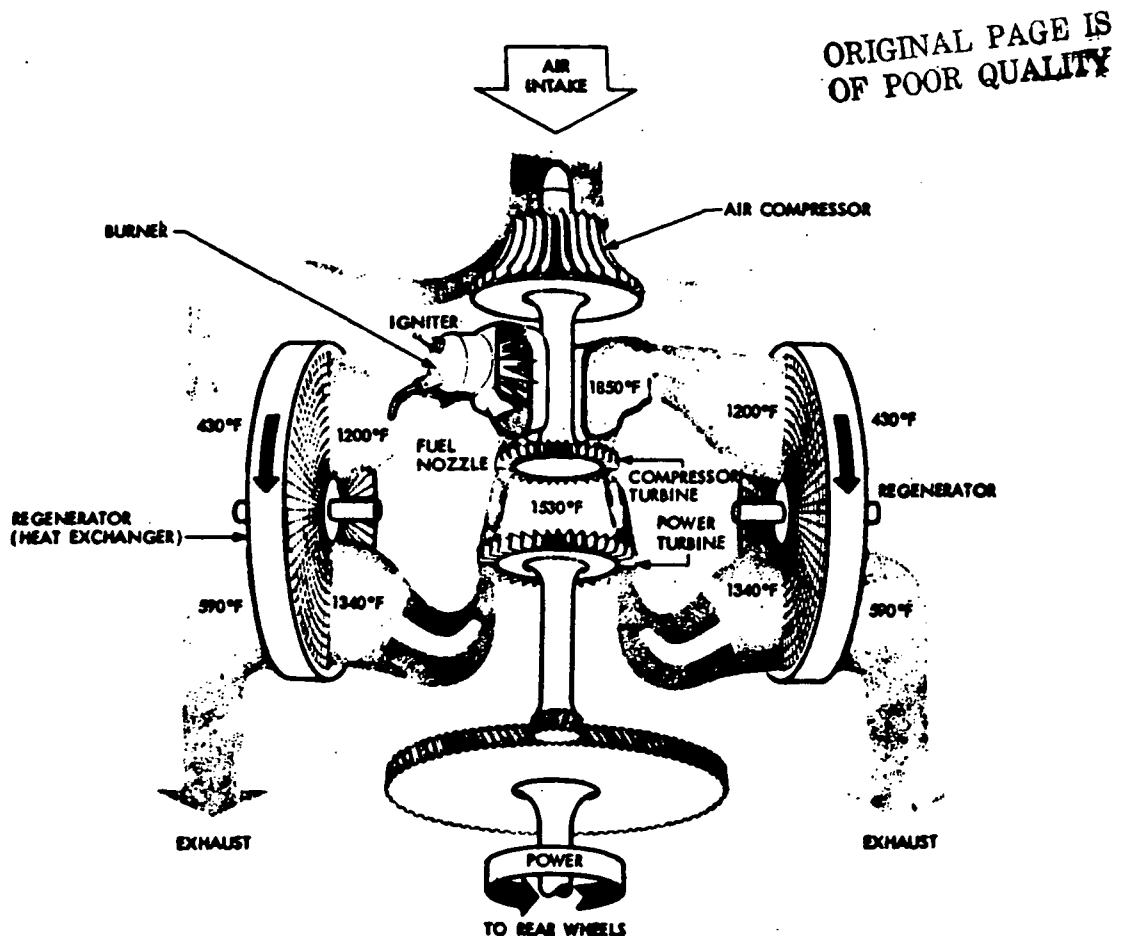


Figure 3-8. Exploded View of the Chrysler Baseline Gas Turbine Engine (Reference 3-15)

Table 3-1. Component Efficiencies, Losses, and Sensitivities
Typical of Non-Size-Critical Regenerative
Automotive Gas Turbines (Reference 3-15)

IGT 404 PM baseline engine performance parameters & sensitivities

Turbine inlet temperature = 1835°F

Brake specific fuel consumption, bsfc = 0.441 lb/bhp-h

Parameter Name	Parameter Value	Parameter Sensitivity (1)
<u>Component Efficiencies</u>		
Compressor	82.4%	+ 0.66
Combustor	99.9%	+ 1.00
Gasifier turbine	87.0%	+ 0.92
Power turbine	89.7%	+ 1.32
Regenerator effectiveness	89.8%	+ 0.92
<u>Pressure Losses ($\Delta P/P$)</u>		
Inlet	1.50%	
Regenerator air side	0.18%	
Combustor & turbine inlet plenum	2.20%	
Inter turbine duct	0.69%	
Turbine diffuser	2.75%	
Regenerator gas side	3.06%	
Exhaust	<u>1.18%</u>	
	11.56%	- 0.80
<u>Leakage and Cooling Flow</u>		
Turbine piston ring leakage	0.17% ⁽²⁾	
Turbine rotor cooling	<u>1.54%</u>	
	1.71%	- 0.62
Overboard leakage (splitlines and main bearing seals)	1.21%	- 0.74
Regenerator leakages, 5.12%		
Rim & crossarm gold seal & disk carryover	1.48%	- 0.70
Crossarm hot seal	1.96%	- 0.60
Rim hot seal	1.48%	- 1.47
Air side bypass	0.20%	- 1.66
Inner muff cooling	1.78%	- 1.66
Block and outer muff cooling	1.78%	- 1.66
Gasifier turbine nozzle cooling	1.00%	- 1.66

Table 3-1. Component Efficiencies, Losses, and Sensitivities
Typical of Non-Size-Critical Regenerative
Automotive Gas Turbines (Reference 3-15)
(Continuation 1)

Parameter Name	Parameter Value	Parameter Sensitivity (1)
<u>Power Losses</u>		
Gasifier driven accessories (fuel pump, oil pump, regenerators)	9.0 hp	- 3.3 hp
Power turbine driven accessories	0.0	
Gear box windage & friction	12.11 hp	- 3.0 hp
Gasifier main bearings	1.72 hp	- 3.3 hp
Power turbine main bearings	1.19 hp	- 3.0 hp
	<u>24.02 hp</u>	
<u>Heat Losses</u>		
Into lubrication system	300 Btu/min	- 405 Btu/min
External	600 Btu/min	- 405 Btu/min
<u>Engine Pressure Ratio</u>	4.0	- 0.34
<u>Engine Cycle Temperature</u>		
Turbine (rotor) inlet	1835°F	+22°F

- (1) Sensitivities shown are in the change in the parameter value to produce a 1.0% reduction in engine bsfc.
(2) Values are percent of supply point airflow.

3.3.1 Compressors

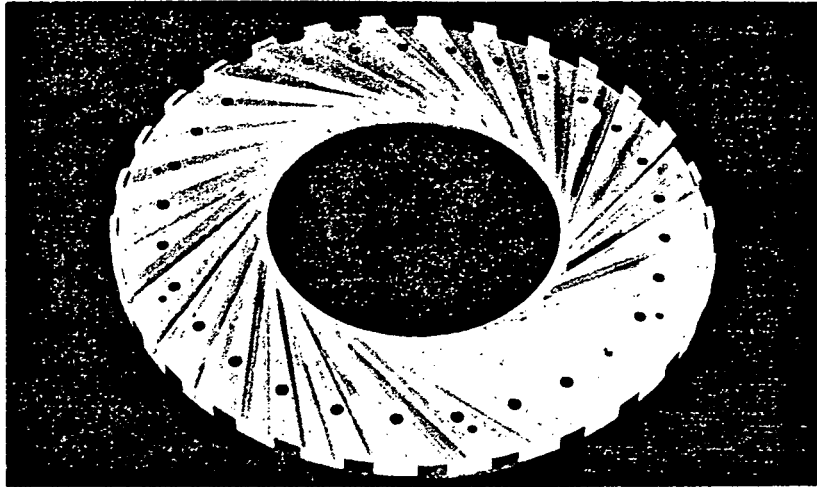
For reasons of simplicity and cost, automotive gas turbine designers have always given preference to the radial type compressor. Initial developments were entirely dependent on spin-off technology from aircraft superchargers with compression ratios on the order of 3 to 3.5 and efficiencies in the mid to high 70s. Considerable progress has been made since the 1950s insofar as the knowledge and optimization of compressor aerodynamics are concerned. High strength aluminum alloys and new production methods have been developed that make possible circumferential velocities on the order of 1500 ft/sec and allow the fabrication of complicated shapes.

The typical design features of a state-of-the-art radial compressor used in today's automotive gas turbines (Figure 3-9) are a one-piece inducer/impeller having swept back and (sometimes) splitter vanes, a vane-less space between the impeller outlet and the diffuser inlet, and the wedge-shaped diffuser vanes that form straight diverging channels of a rectangular cross-section. The diffuser usually discharges into a large scroll or plenum chamber. For flow rates of interest for automotive gas turbines (0.8 - 2.0 lb/sec), compressors of this type are capable of producing a compression ratio up to about 6 at efficiencies in the high 70s. Peak efficiency of over 80 percent can be maintained up to a compression ratio of about 4. Compression ratios up to 12 are theoretically possible, but efficiencies drop off markedly with increased speeds, and the usable range between surge and choked flow becomes unfavorably narrow (Figures 3-10 and 3-11). Supersonic flow at the diffuser inlet and a high sensitivity to changes in angle of attack are primarily responsible for this drop in efficiency.

Further improvement in peak compressor efficiency is possible, but it is expected to be small and not of a magnitude that will improve the fuel economy and transient behavior of present state-of-the-art engines significantly. R&D approaches to solving the problem of maintaining high compressor efficiency over the urban driving cycle include the development of variable diffusers that allow for a substantial map-shift and efforts to make compressors less sensitive to off-design operation (i.e., flatter characteristics). A proven approach to achieve flatter characteristics is to widen the vane-less space; this reduces the Mach number at the diffuser inlet and tends to make the flow near the leading edge of the diffuser less sensitive to angle-of-attack variations (larger radii).

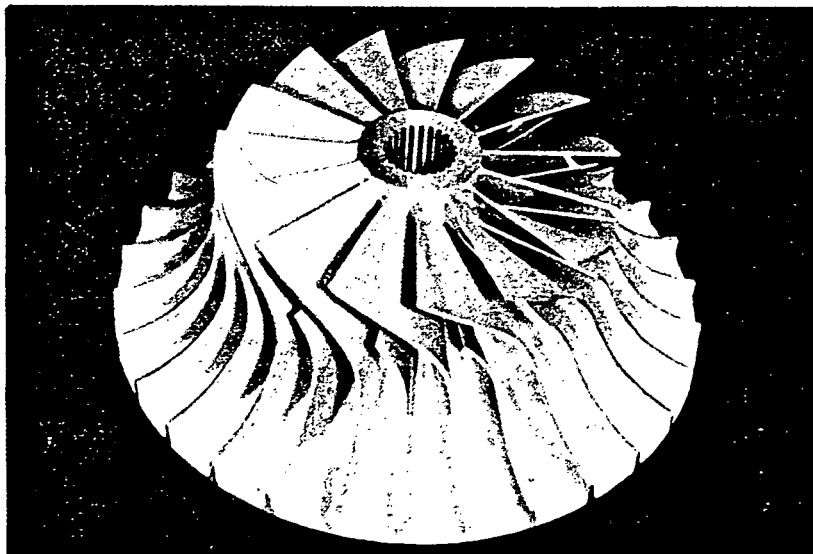
The design of the diverging portion of the diffuser and its optimization have been extensively addressed in R&D work over the years, giving preference to a rectangular channel design. In order to prevent excessive boundary layer buildup and separation, optimum diffusers require a slender two-dimensional divergence and a radial extension of about one throat radius from the compressor axis, which, unfortunately, is not always feasible in automotive gas turbines because of packaging constraints. Diffusers diverging between parallel planes are frequently the preferred choice from the cost standpoint, but they are definitely less efficient because of a large increase in surface area and boundary layer thickness.

Because of their advantages from the standpoint of boundary layer growth (minimum surface area), there is renewed interest in conical diffusers. Conical diffusers (Figure 3-12) are not new, but they were usually rejected in the past by engine developers because of production cost. The intersecting portions of the cones forming the diffuser lip require a high degree of accuracy and have to be machined. Today's casting methods are sufficiently accurate to produce conical diffusers that satisfy aerodynamic requirements and cost demands as well.



DIFFUSER

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IMPELLER

Figure 3-9. Typical Configuration of a State-of-the-Art High-Performance Radial Compressor (from Reference 3-2, The Gas Turbine Engine by Jan P. Norbye. Copyright 1975 by the author. Reprinted with the permission of the publisher, Chilton Book Co., Radnor, PA.)

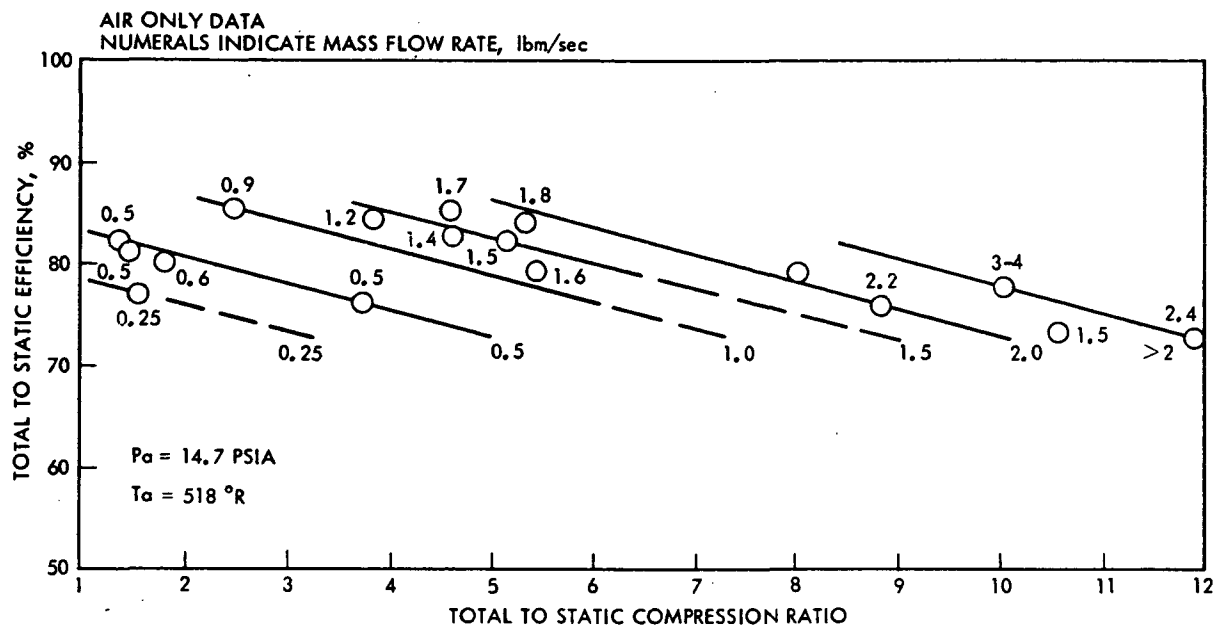


Figure 3-10. State-of-the-Art Centrifugal Compressor Performance, ca. 1975 (Reference 3-17)

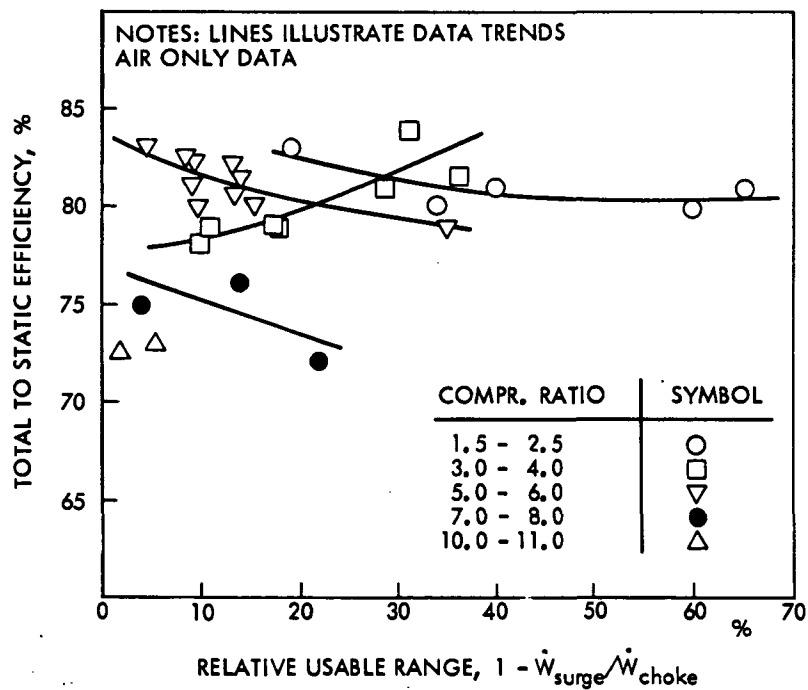


Figure 3-11. State-of-the-Art Centrifugal Compressor Efficiency Vs. Usable Range, ca. 1975 (Reference 3-17)

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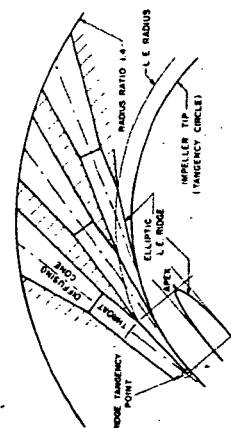
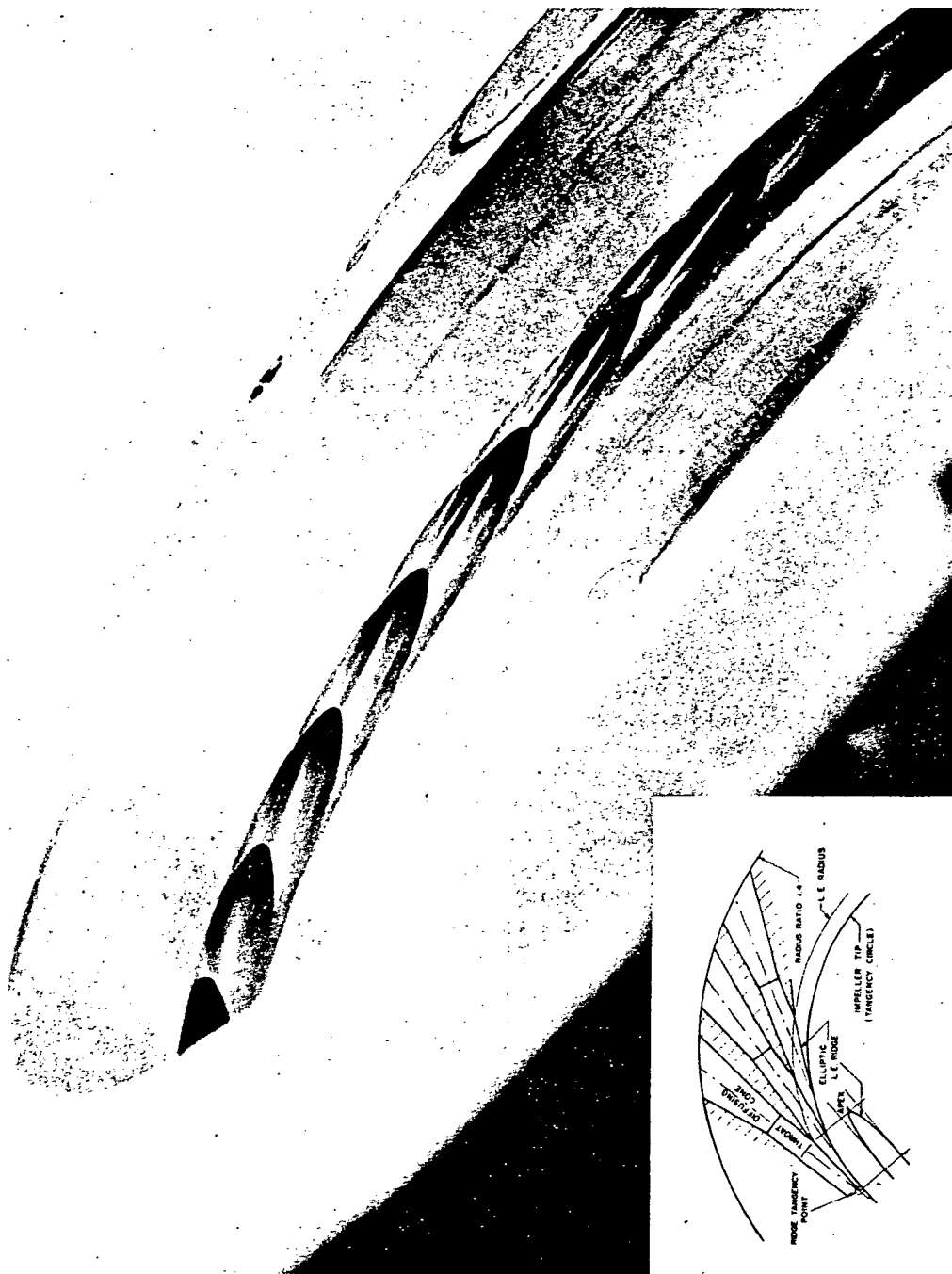


Figure 3-12. Typical Conical Diffusor (Reference 3-9)

The potential for the optimization of a spacially restricted diffuser has not been utilized in many automotive turbine designs. In situations where the radial extension of the diffuser is restricted, as in the case of the Upgraded Chrysler/ERDA engine, a rapidly diverging diffuser that discharges directly against a surface (Figure 3-13A) is an efficient approach to the problem. A relatively large equivalent divergence of up to about 18 degrees is possible without efficiency penalties for an optimum relationship between the exit radius (r) and standoff (δ) between the diffuser exit and the impinger surface. This approach might have been a more effective solution to the space problem in the Upgraded Chrysler engine than the elbow design chosen.

3.3.2 Turbines

The axial flow turbine is the preferred choice of most engine designers for two- (and three-) shaft engines. Some developers have, however, chosen a radial turbine for the gasifier even though this leads to a longer engine. For single-shaft engines, the radial turbine is preferred, although it is not the only choice. Both axial and radial turbines have their pros and cons. The rationale for the choice of one type over the other must include both performance characteristics and cost. The efficiency of the radial turbine (Figure 3-14) reaches its maximum at a lower specific speed, which is an advantage from the part-load economy standpoint. The ratio of total to static efficiency that can be achieved with a radial turbine is higher than that of an axial turbine unless a highly efficient exhaust diffuser can be provided, which is usually not possible in automotive applications. Radial turbines are capable of expansion ratios up to 6:1. It takes two to three axial turbine stages to accomplish the same expansion under conditions that are acceptable from stress and efficiency standpoints. A strong advantage of the axial flow turbine over the radial type (Figure 3-15) is its smaller size and volume. The inertia of an axial turbine is only a fraction of that of a radial turbine and this has a strong effect on engine response. Failure probability considerations, smaller volume, and more favorable stress-distribution make the axial turbine more adaptable to the use of ceramic materials than the radial type.

As shown in Figure 3-16, the inlet temperature of turbines has increased considerably over the last 20 years and has reached a point (1950°F) at which the potential for the application of superalloys is exhausted. Current state-of-the-art air cooling techniques allow turbine temperatures of about 2300°F. Air cooling has usually been rejected in automotive gas turbine developments because of the size, complexity, cost, and aerodynamical penalties associated with its use (thicker trailing edges, generally plump blade profiles, mass flow losses). This, however, does not rule out air cooling as a workable approach for automotive gas turbines. Ceramic coatings have been developed for cooled rocket nozzles and aircraft gas turbine components which, if applied to automotive engines, would reduce the amounts of air needed for air cooling by a factor of two or more, and reduce the surface temperature considerably, up to 380°F (130°C).

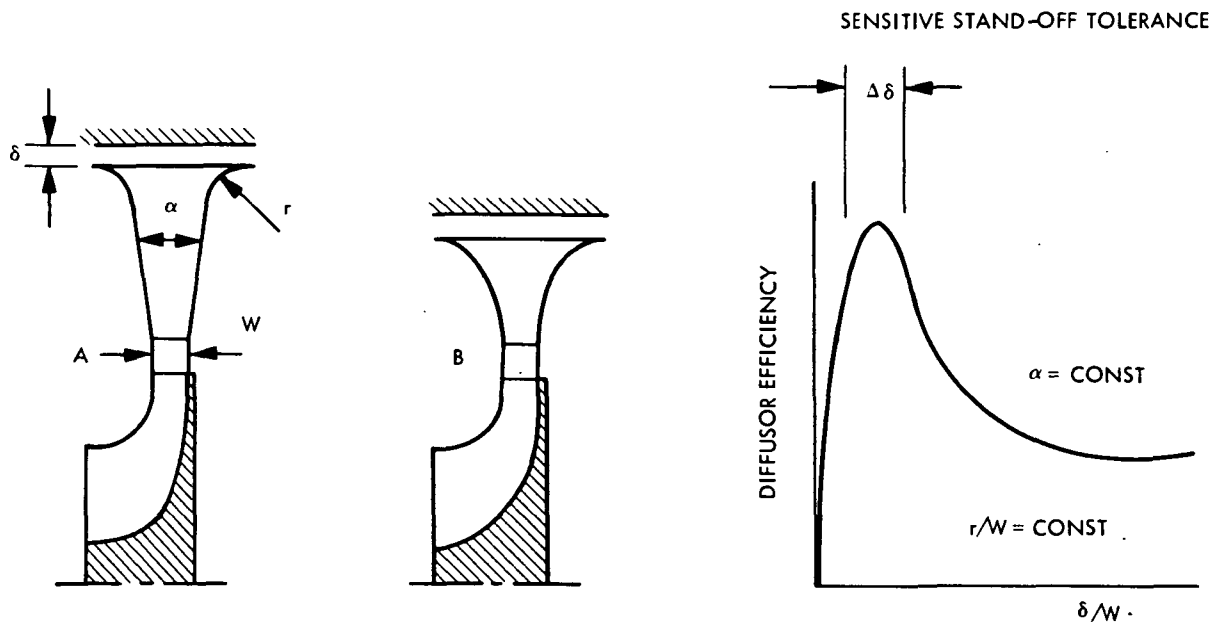


Figure 3-13. Efficient Short Diffuser Concepts. A-Linear Divergence, B-Exponential Divergence

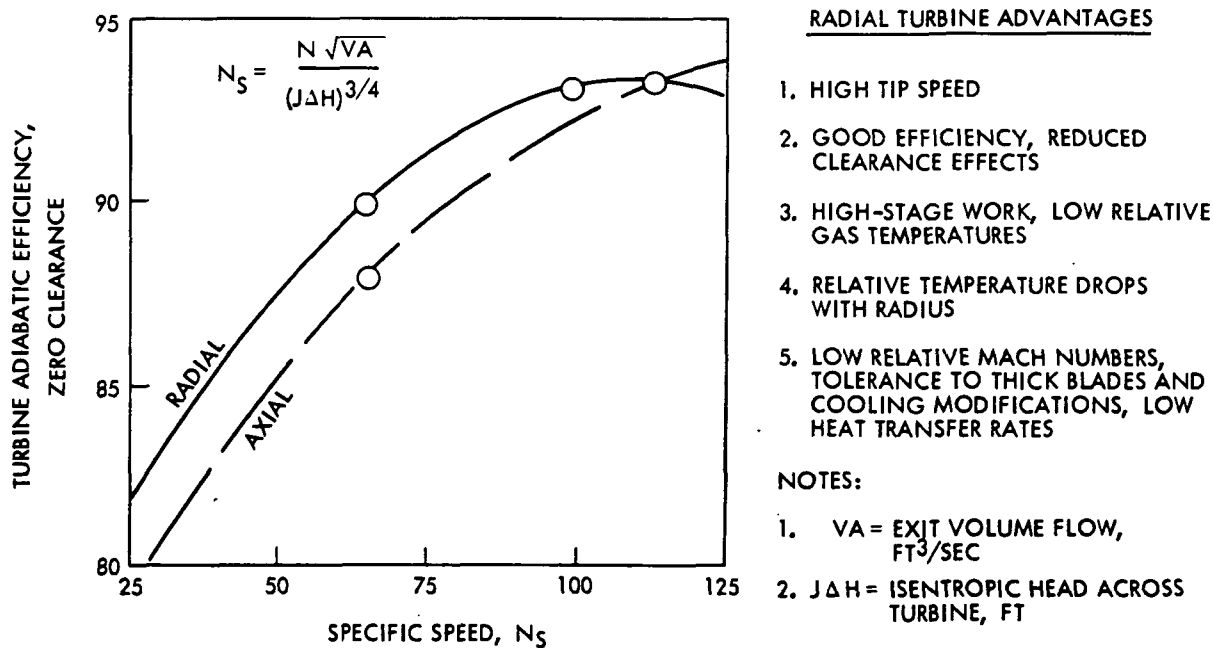


Figure 3-14. Axial and Radial Turbine Performance (Small Turbines) (Reference 3-8)

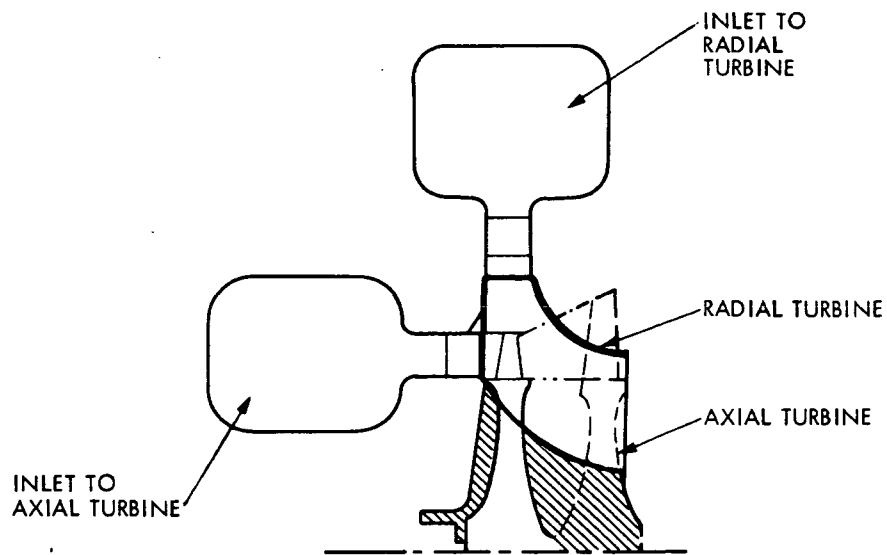


Figure 3-15. Size Comparison Between Axial and Radial Turbines for Equal Flow Rates

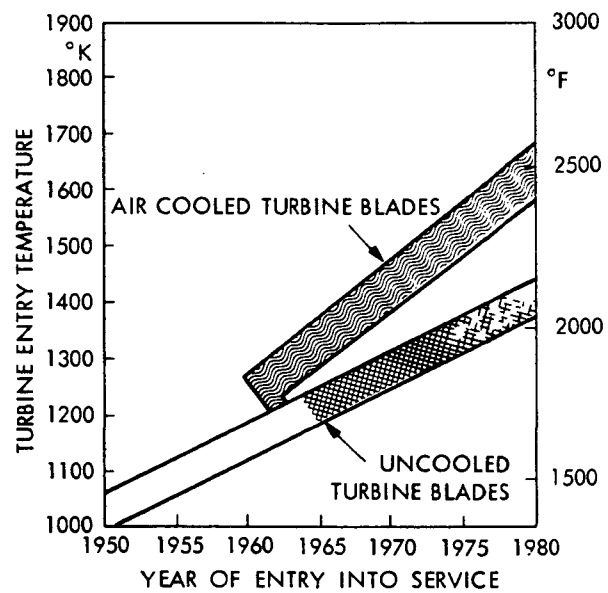


Figure 3-16. Increase in Turbine Inlet Temperature During the Past 20 Years and as Projected for the Future (Based on Reference 3-18)

The AiResearch Company has proposed a unique approach to the channeling problem with small air cooled turbines which is estimated to be workable for inlet temperatures of up to 2300°F. The proposed turbine is a radial inflow type and consists of a stack of thin walled laminates with interspaces between them for the cooling air, which enters the turbine in the center and exits at the surfaces interfacing with the hot gas. The air channels are producible in a cost-effective way by chem-milling or photo-etching the individual layers before they are joined together and machined to size.

Although it appears that air cooled turbines have a potential for high temperature application in automotive gas turbines, present planning and research is directed toward the development of ceramic turbine materials, with silicon nitride and silicon carbide being the most promising candidates. As shown in Figure 3-17, these materials are expected to make continuous operation at turbine inlet temperatures on the order of 2500°F (1370°C) possible. Ceramics are also expected to lower the cost of turbines considerably.

The primary problem with ceramics is their inherent brittleness which makes them very sensitive to impurities, surface damage (scratches) and other stress risers introduced by thermal gradients and mechanical loads. The use of compliant layers or porous layers of reaction sintered silicon nitride at ceramic-metal interfaces, are to date the best approaches to minimize stresses. As shown schematically in Figure 3-18, the introduction of ceramics is not only a matter of replacing existing superalloys. It means developing an entirely new technology. Government-funded programs that pursue the development and the implementation of candidate ceramic materials are in progress at Ford, AiResearch and the Detroit Diesel Allison Division of General Motors. Similar efforts are also being made in Japan and in Europe. Details pertaining to these programs and the development status of ceramics are discussed in Section 9 of this report.

3.3.3 Combustors

Before carbon monoxide and nitrogen oxide emissions became a matter of concern, the combustors of automotive gas turbines were of the simple flame diffusion type. These combustors consist of an array of fuel injection nozzles and a fixed-geometry arrangement of flame holders in the wake of which near-stoichiometric combustion takes place in the primary zone. The combustion gases are then diluted with bypassed air through perforated or slotted walls and cooled to a temperature compatible with the turbine materials. The main problems associated with the development of diffusion burners were combustion stability, temperature distribution, hot spots, and the achievement of a low pressure drop at a high combustion efficiency. The preferred configuration in most automotive gas turbines was the reversed-flow can-type combustor although straight through-flow barrel and toroidal chambers have been used (Reference 3-2, p. 38).

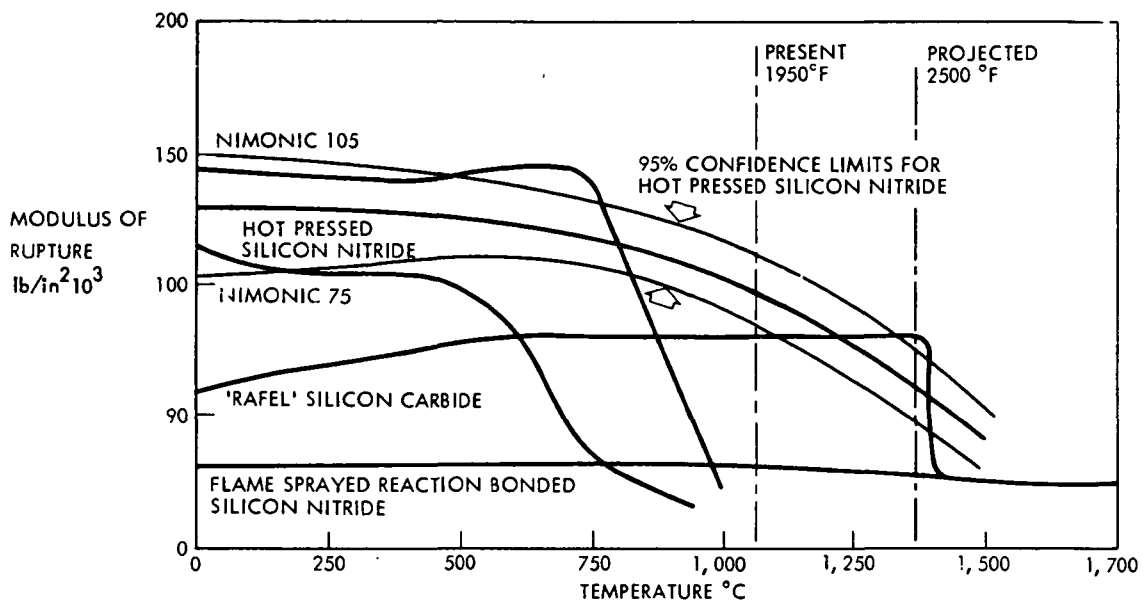


Figure 3-17. Effect of Turbine Temperature on Strength of Materials

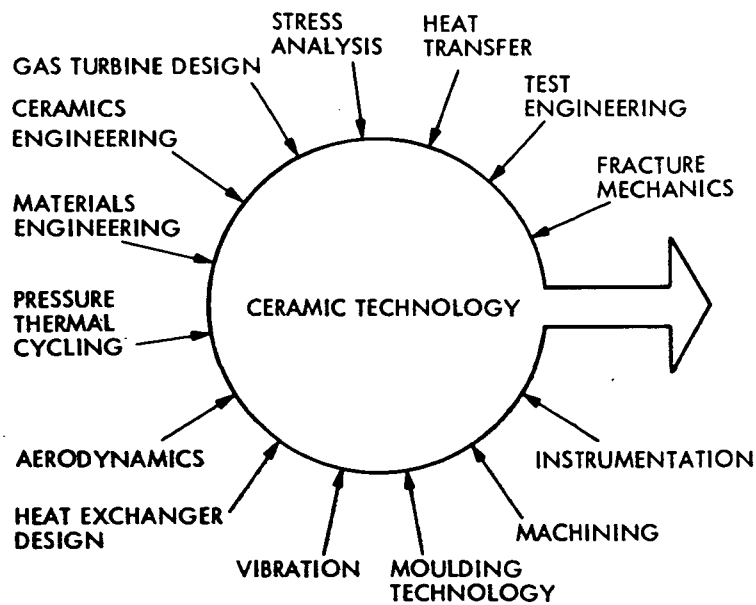


Figure 3-18. Approaches to the Development of Ceramics Design and Production Technology

Diffusion flame combustors have been developed to a high degree of efficiency. Their emission of unburned hydrocarbons (HC) is low, but they produce amounts of carbon monoxide (CO) and oxides of nitrogen (NO_x) that are not compatible with today's emission standards. Droplet burning, a very short dwell time in the reaction zone, and the dilution of the hot gases with air are mainly responsible for this. As Figure 3-19 shows, a reduction of CO and NO_x emissions of about 90 percent was required in order to meet today's emission standards for CO and NO_x . To keep NO_x low, the highest combustion temperature produced must remain well below 3000°F , and the dwell time in the reaction zone be as short as possible. To control CO, sufficient dwell time in the reaction zone must be provided. The problem is to find a good compromise between these factors and to keep the mixture ratio within the narrow limits necessary to assure that neither of the pollutants exceeds the prescribed maximum. This is particularly difficult during engine start, acceleration, and deceleration transients.

Evaporative combustion seems to be an attractive approach to meeting the multitude of requirements imposed on automotive gas turbines in regard to efficiency and emission standards. One system that is successfully applied in small gas turbines, other than automotive, is schematically shown in Figure 3-20. Here the fuel is injected

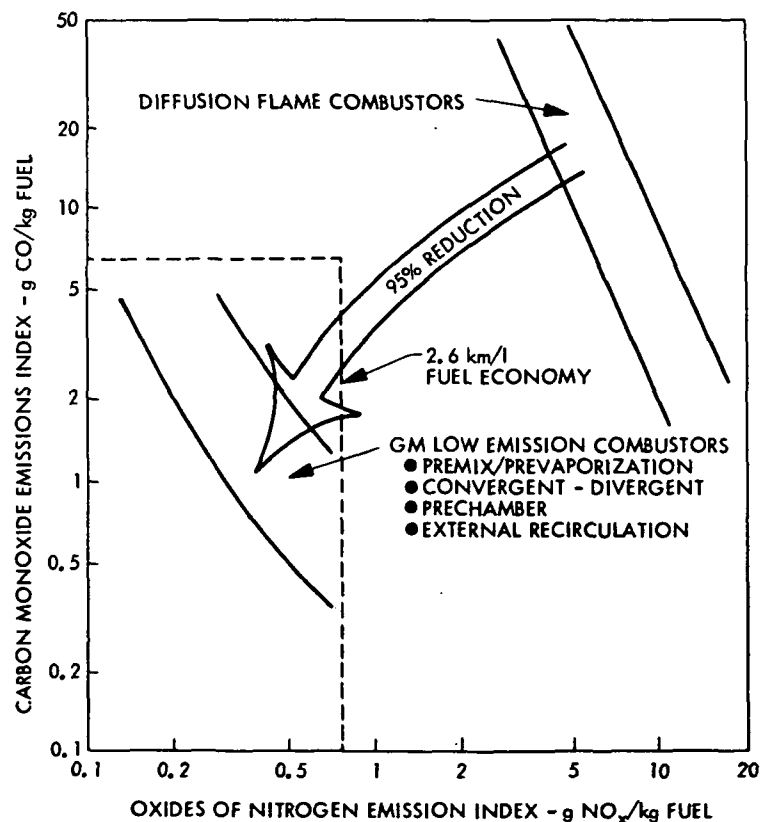


Figure 3-19. 1978 Emission Cleanliness Objectives for Light Duty Vehicles (Reference 3-19)

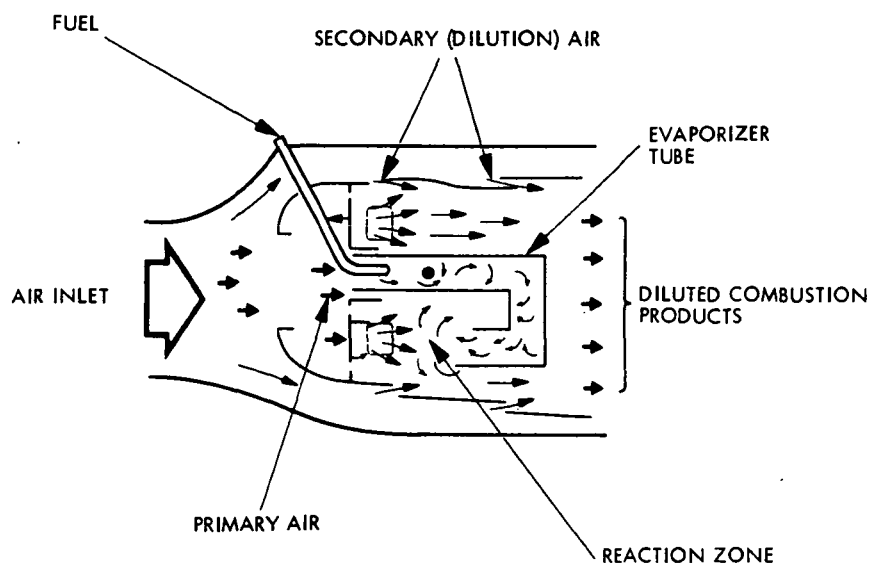


Figure 3-20. Schematic of Evaporative Combustion Chamber (Reference 3-20)

into a U-shaped tube (that protrudes into the combustion zone) where it is mixed with a small portion of air, and is heated and vaporized. The vaporized mixture exits the tube and enters the mainstream in the wake of a flame holder where it reacts quickly with a minimum of delay. The advantages of the system are a low fuel injection and system pressure, relatively large nozzle orifices, a uniform temperature distribution, short high temperature dwell time (low NO_x) and, as shown in Figure 3-21, high combustion efficiency, good tolerance to the use of different fuels, generally flat characteristics, and a potential for extremely lean combustion, which is desirable from the NO_x standpoint.

One approach to inhibiting the formation of NO_x is extremely lean burning of a premixed, prevaporized charge below the NO_x critical temperature. The feasibility of this approach was demonstrated in 1972 by General Motors (Reference 3-17). A fixed-geometry burner that had no flame holders was used; an additional air pump was required to accomplish premixing. The emission levels achieved were very low, but the system exhibited an unacceptably narrow range between the mixture ratio that would produce a flame-out and the ratio at which the NO_x level would start to rise steeply. Variable-geometry features seemed to be needed to solve this problem. Approaches in that direction were made but were plagued with flashback, unreliability, and mechanical problems.

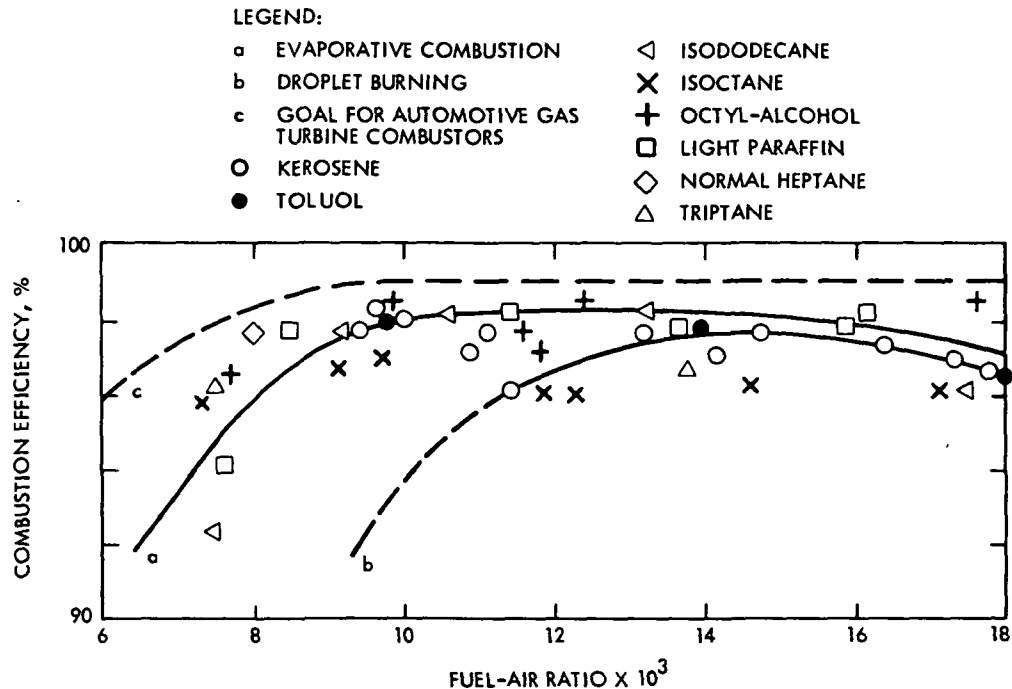


Figure 3-21. Comparison of Typical Combustor Characteristics (Reference 3-20)

Figure 3-22 shows three low-emissions combustor concepts which use the premixed, prevaporized approach. The dual stage combustor (Figure 3-22A) provides fuel injection into two different zones. The fuel split between the two stages of combustion can be varied to obtain a wider range of low-emissions operation than the simple premixed combustor. The Solar combustor (Figure 3-22B) uses variable geometry to change the ratio of primary air flow to dilution air flow as engine speed changes. Premixing tubes supply the fuel-air mixture into the rear of the combustion chamber for mixing and recirculation in the primary combustion zone.

A simpler design is that of Chrysler's advanced combustor, Figure 3-22C. This design employs fixed geometry with a single premixed stage. Combustion stabilization is provided by a continuously burning torch that discharges tangentially into a larger chamber where lean-burning and further dilution to turbine inlet temperature takes place. One advantage of this concept is that the premixed stage can be completely shut down while only the torch remains lit during engine deceleration.

Emissions from the dual stage and Solar concepts are compared with the Baseline Chrysler diffusion-flame combustor at steady-state conditions in Figure 3-23. The advanced Chrysler combustor of Figure 3-22C has been demonstrated to have CO emissions near the level of the dual stage combustor and NO_x emissions below those of the dual stage combustor.

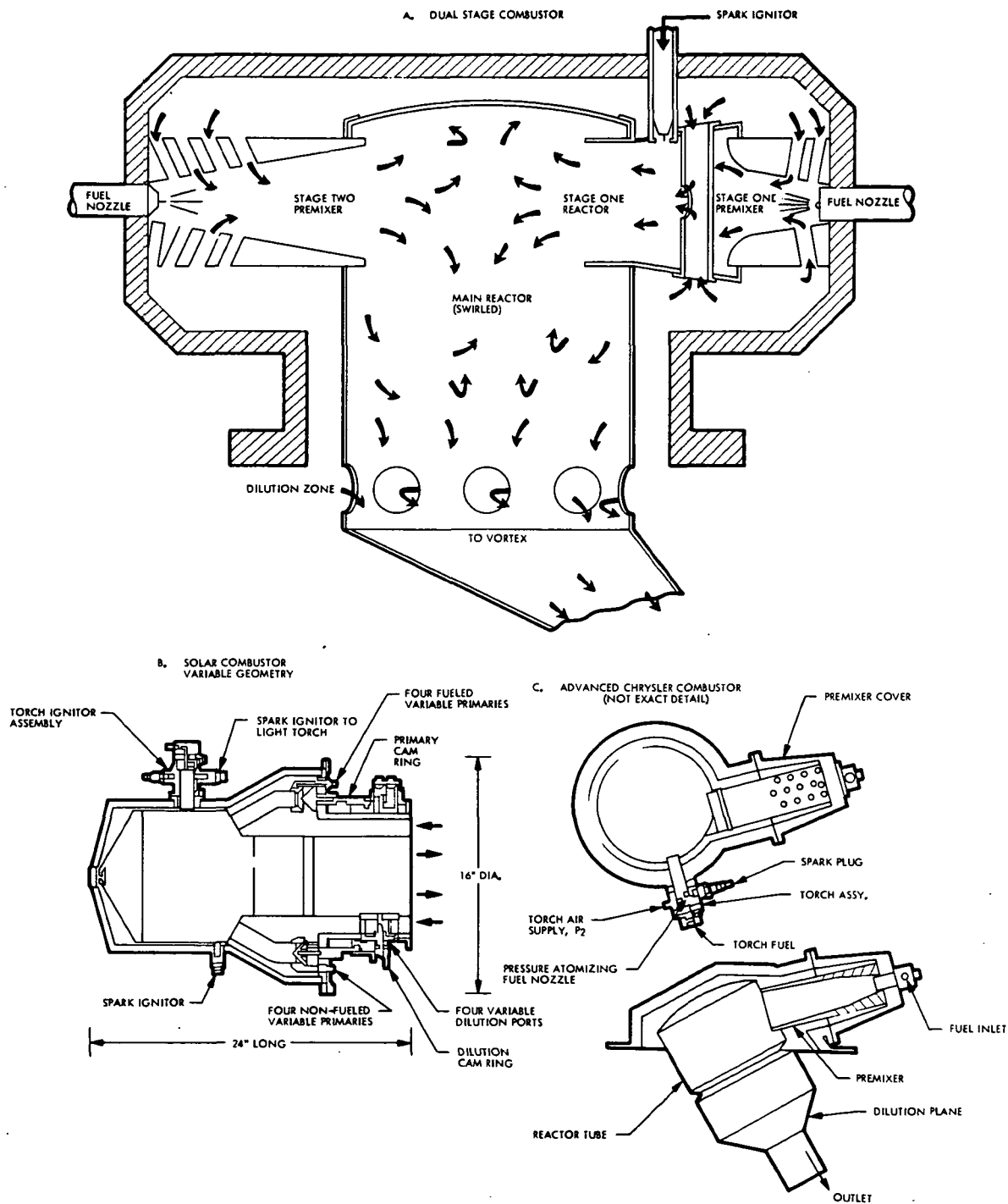


Figure 3-22. Typical Low Emissions Combustor Concepts
(Reference 3-21)

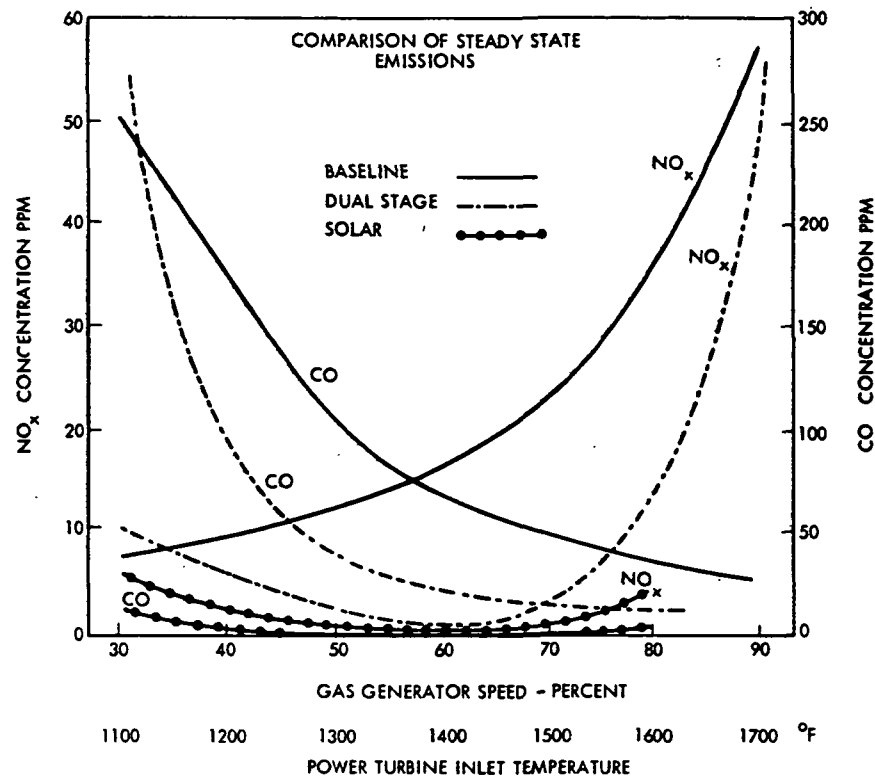


Figure 3-23. Effect of Burner Type and Temperature on Engine Emissions (Reference 3-21)

Efforts to introduce high temperature ceramics into the hot areas of the combustor are in progress. Current problems with ceramics in the burner area include the development of cracks due to temperature cycling and thermal gradients, particularly where stress concentrations are unavoidable such as dilution holes. Alternate design approaches supported by finite element analysis may solve these problems in the near future.

Multifuel capability is an essential asset of the gas turbine. It is not clear at this time whether a burner system that will operate satisfactorily with different fuels interchangeably and simultaneously meet emission standards can be developed. The development of interchangeable or revolving burner heads may be needed.

Development work with advanced aircraft gas turbine burners that may also be applicable in future automotive gas turbines is in progress at the NASA Lewis Research Center (Reference 3-22). The swirl can combustor shown in Figure 3-24A divides the primary combustion zone into a large number of smaller flames to facilitate premixing and lean burning and to reduce residence time. A NO_x reduction of 30 to 33 percent has been demonstrated with chambers of this design. Two additional

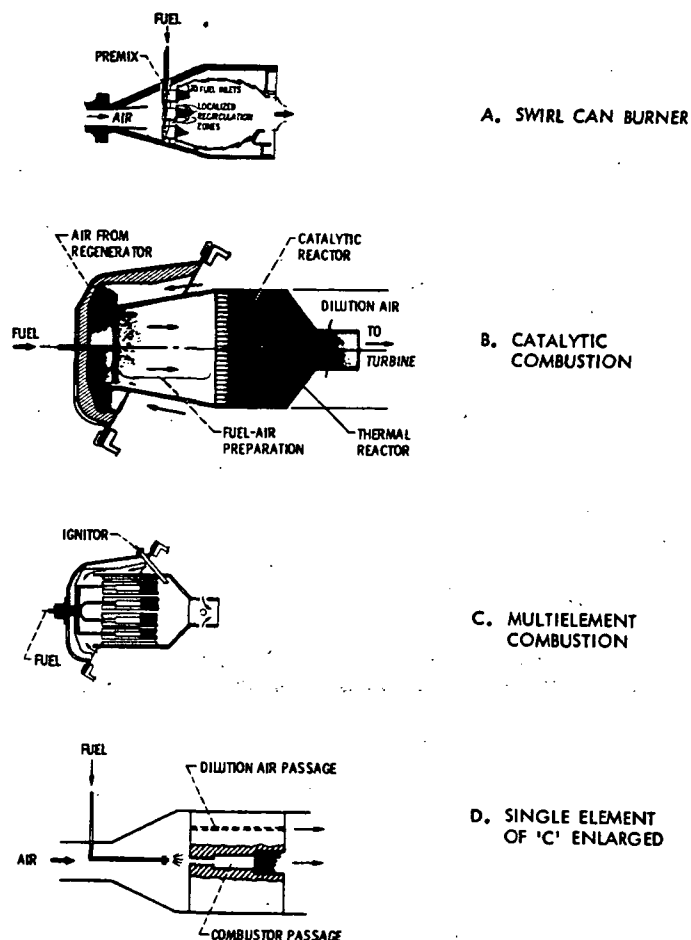


Figure 3-24. Advanced Burner Concepts
(Reference 3-22)

concepts under investigation at the NASA Lewis Research Center are the catalytic and the multielement combustor. In the catalytic combustor (Figure 3-24B) vaporized fuel is reacted in a catalyst bed which allows combustion to take place at very lean mixture ratios that are well below the flammability limits and at a temperature that is only slightly higher than the turbine inlet temperature (Reference 3-23). Low NO_x is possible at high combustion efficiencies with this burner approach. A disadvantage is that the catalyst bed has to be reheated for cold start.

In the multielement combustor (Figure 3-24D) the hot zone of the flame is conductively cooled by means of the diluting air across parallel ceramic walls which separate the flame and the diluting air from each other until they mix downstream. The parallel passages are integrally cast into a ceramic block and there is sufficient heat exchange across the channel walls to prevent the adiabatic flame temperature from being reached. Encouraging results in regard to reduced flame temperature and residence times have been obtained.

3.3.4 Heat Exchangers

The practicable means of efficiently recovering heat from the turbine exhaust in an automotive gas turbine are the fixed boundary counterflow recuperator and the rotary heat exchanger, also referred to as a regenerator (Figures 3-25 and 3-26). Both have their pros and cons. The rotary regenerator is most efficient, over 90 percent (Figure 3-27), because small hydraulic radii on the order of 1/2 mm (0.020 inch) are feasible at an acceptable pressure drop of 3 to 5 percent on the hot gas side and less than 1 percent on the air side (Figure 3-28). In addition to efficiency, a major advantage of the rotary heat exchanger is that it is self-cleaning. The buildup of deposits is inhibited by the periodic reversal of the core flow and the alternate exposure of deposits to a highly oxidizing environment. Disadvantages of the rotary regenerator are the losses due to air leakage and carryover, which are about 5 percent for a compression ratio of 5 (Figure 3-29). Further disadvantages are the mechanical complexity required for the seals, bearings, and gears necessary to align and drive the core, which rotates at 15 to 20 RPM.

The primary advantage of the recuperator is that there are no moving parts and no leakage. Unfortunately, its efficiency is 5 to 10 percent lower than that of a regenerator of equal volume. A higher compression ratio (~ 7) is required to obtain acceptable cycle efficiencies. Another major disadvantage of the recuperator is that it tends to contaminate. Considerably larger flow passages and hydraulic radii are required to avoid clogging and to obtain a tolerable pressure drop. More surface area, volume, and structural weight has to be provided to transfer a given quantity of heat. Experience with recuperators in automotive gas turbines has also shown that they are sensitive to temperature variations and can develop fatigue cracks.

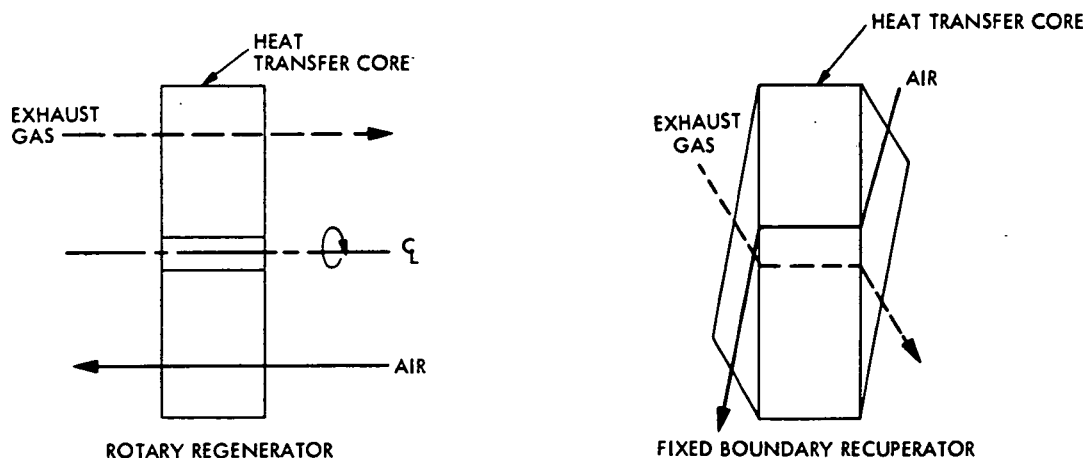
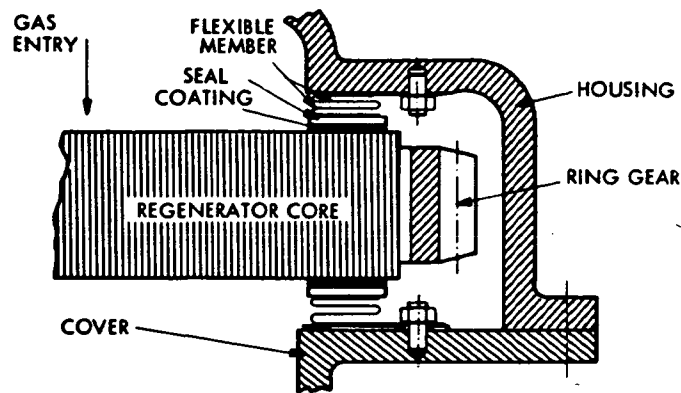


Figure 3-25. Flow Schematic of Competing Heat-Recovery Systems for Automotive Gas Turbines



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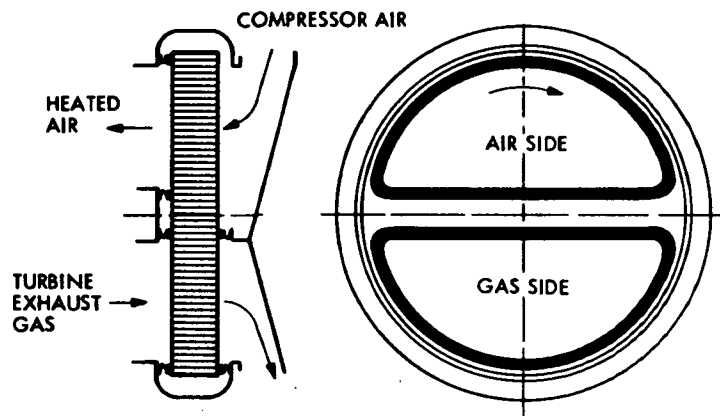


Figure 3-26. Structural Concept of Typical Rotary Disc Type Regenerative Heat Exchanger

Cross- and counterflow recuperators as well as drum and disc type rotary regenerators have been used in developmental engines. Because of its high efficiency, particularly at part load (Figure 3-27), the disc-type rotary regenerator has emerged as the preferred choice. The development of rotary heat exchangers has been a pacing factor in automotive gas turbine programs due to problems of leakage, seal wear, and durability but definite progress is being made (Reference 3-24). A variety of approaches have been taken to resolve the seal and the core drive problems. Flexible and rigid seals and rim and hub-driven core designs are still being evaluated. The development of ceramic cores (Reference 3-24) has greatly reduced problems with core bowing, warpage, and distortion, which were the primary reasons for the excessive wear of seals and drive mechanisms encountered with metal cores. A detailed discussion of ceramic regenerator development is given in Section 8-4.

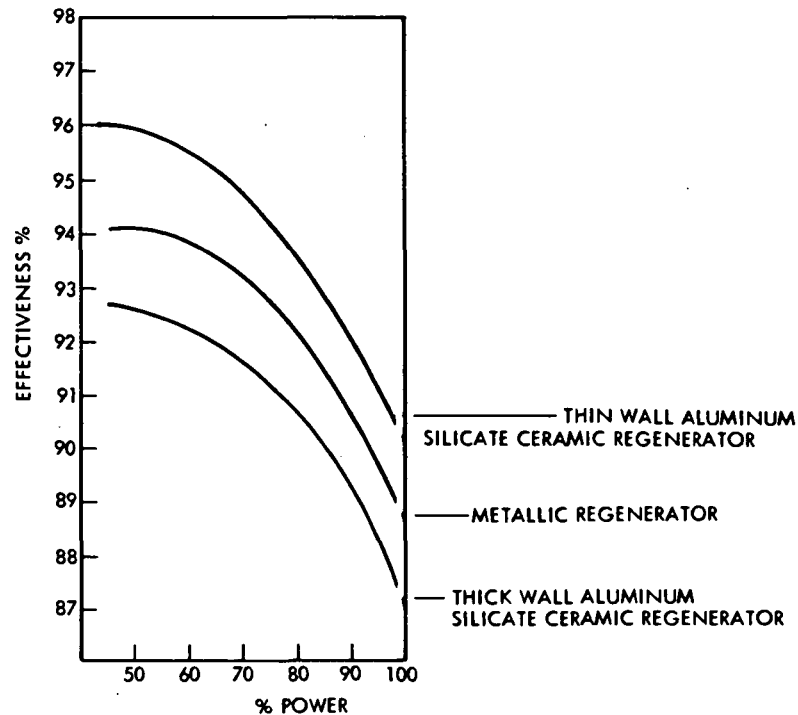


Figure 3-27. Part-Load Behavior and Efficiency Potential of Disc-Type Rotary Heat Exchangers

Some developers believe that the leakage and the mechanical reliability problems with rotary regenerators will be very difficult to completely solve and are still doing developmental work on recuperative heat exchangers. Based upon their background with stacked plate type counterflow recuperators (Figure 3-30), the AiResearch-Garrett Corp. has shown a preference for the use of recuperators for passenger car turbines and has chosen recuperators for their 601 truck-engine now under development. Solving the contamination problem of recuperators might require the development of suitable fuel additives that inhibit the formation of deposits or the development of cost-effective and practicable cleaning methods unless the temperatures of all inner surfaces are kept sufficiently high to effect an oxidation of hot and cold side deposits.

3.3.5 Variable-Geometry Compressors and Turbines

The use of variable guide-vanes in the hot part of the automotive gas turbine is almost unavoidable. Variable guide-vanes were first introduced in two-shaft engines for the power turbine in order to control heat exchanger temperature, increase turbine torque output, and for braking (Figures 3-31, 3-32, and 3-33). The design concept most commonly applied (Figure 3-34) consists of collectively driven vanes that are pivoted about a radial axis by means of segmented gears that

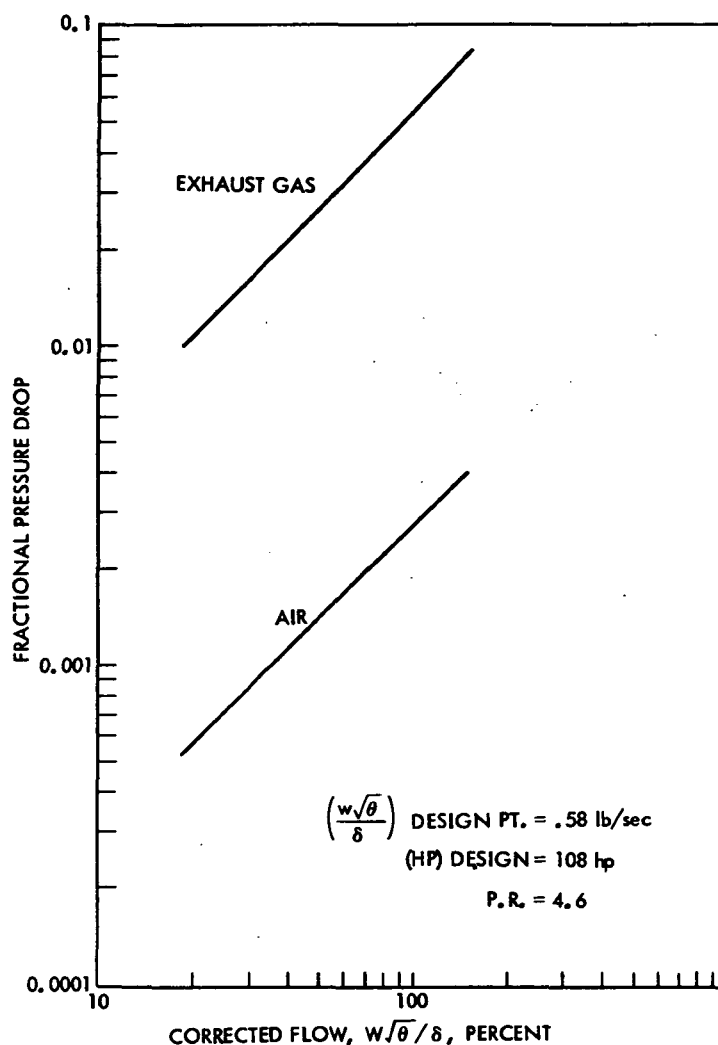


Figure 3-28. Rotary-Ceramic Regenerator Off-Design Performance (Reference 3-8)

engage into a common ring gear driven by a hydraulic actuator. The inner and outer walls of the flow path swept by the vanes are spherically shaped to maintain minimum radial clearance during vane rotation except for power braking which requires a 90-degree reversal of the vanes. Figure 3-35 shows a typical design approach for a variable-geometry radial turbine.

Variable guide-vanes at the compressor inlet have been introduced more recently to increase compressor aerodynamic work for power augmentation during vehicle acceleration. This allows the designer to reduce the installed minimum engine power required for acceleration and thus improve vehicle part-load fuel economy. Preferred design concepts are shown in Figure 3-36. Configuration B represents the concept used in the Chrysler Upgraded engine. It is of the flap type where only

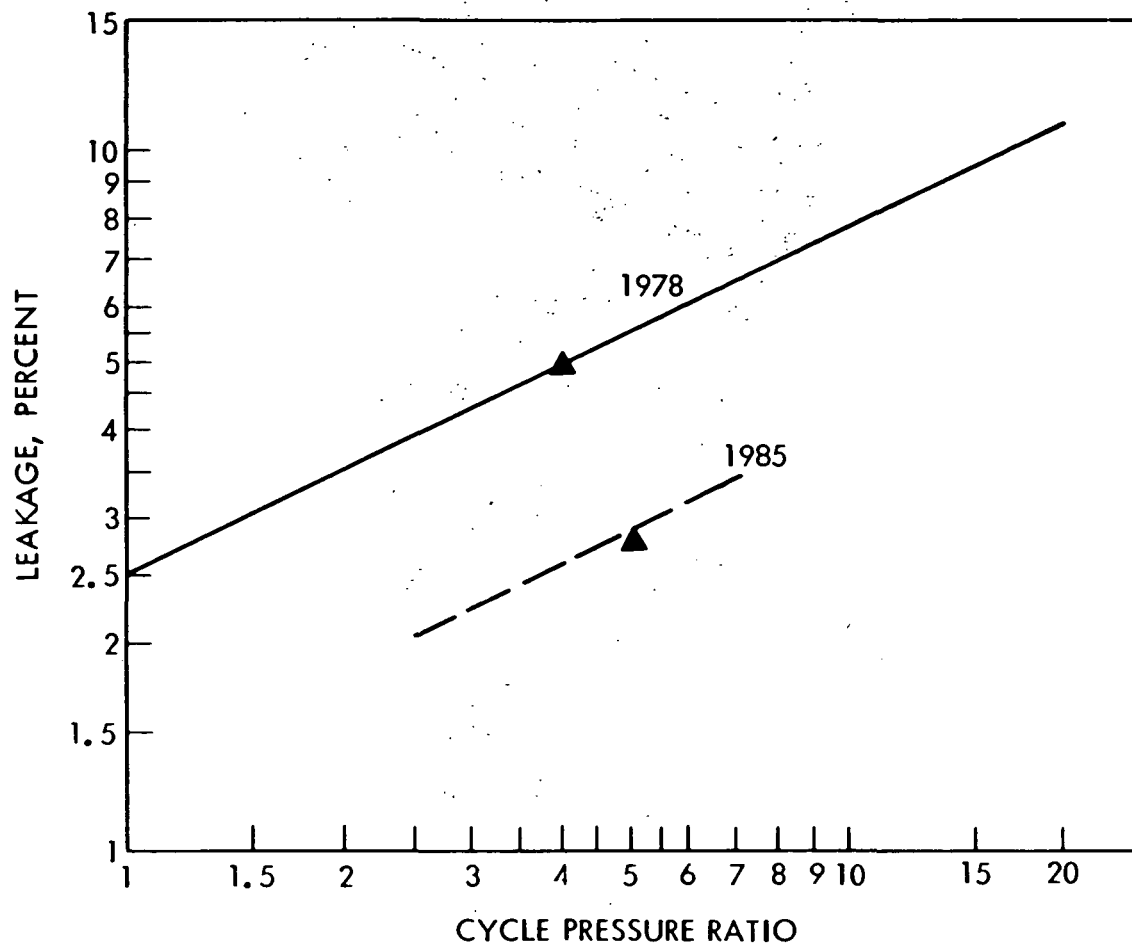


Figure 3-29. Regenerator Seal Leakage Design Goals (Reference 3-8)

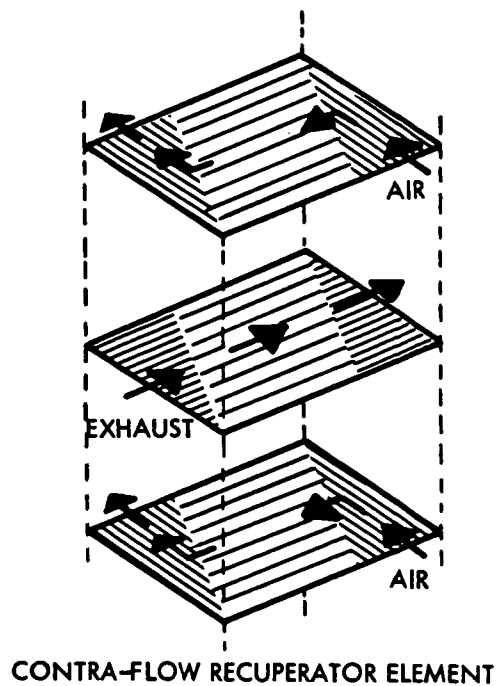
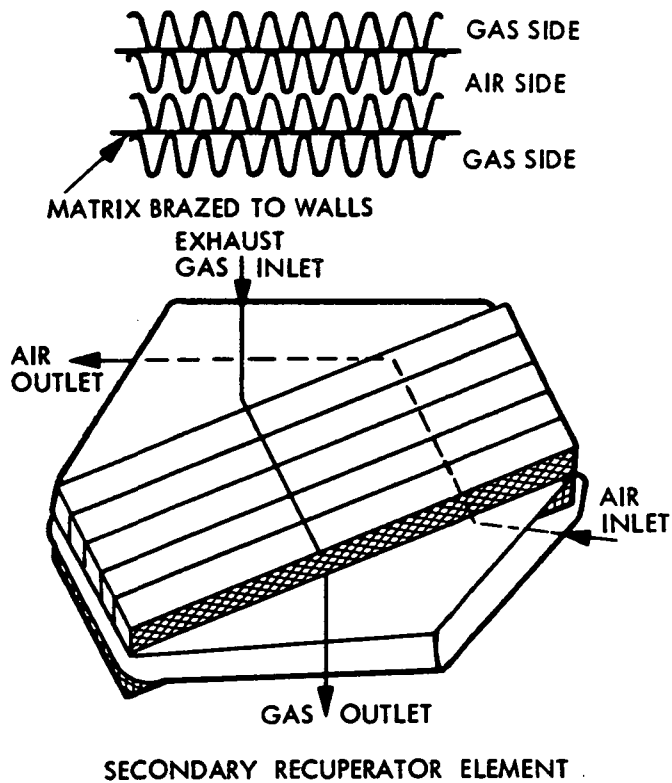


Figure 3-30. Schematic of State-of-the-Art Recuperative Heat Exchanger (Reference 3-8)

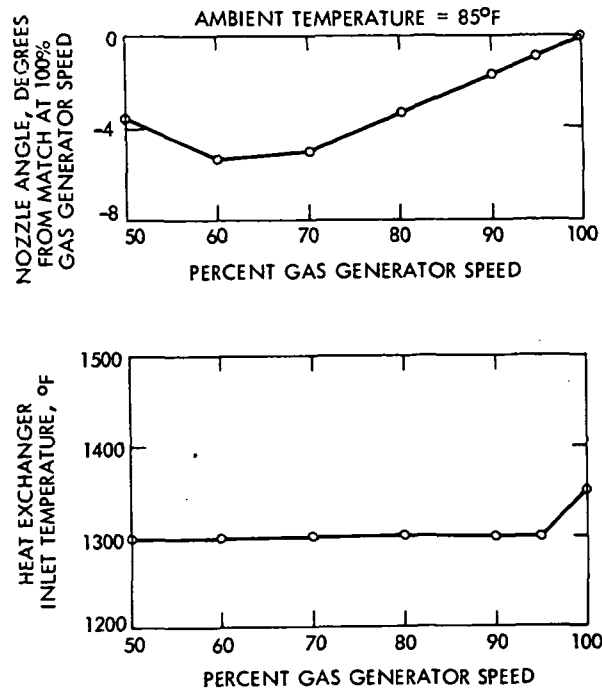


Figure 3-31. Change of Power Turbine Nozzle Angle Required for Constant Heat Exchanger Inlet Temperature, T_8 (Reference 3-25)

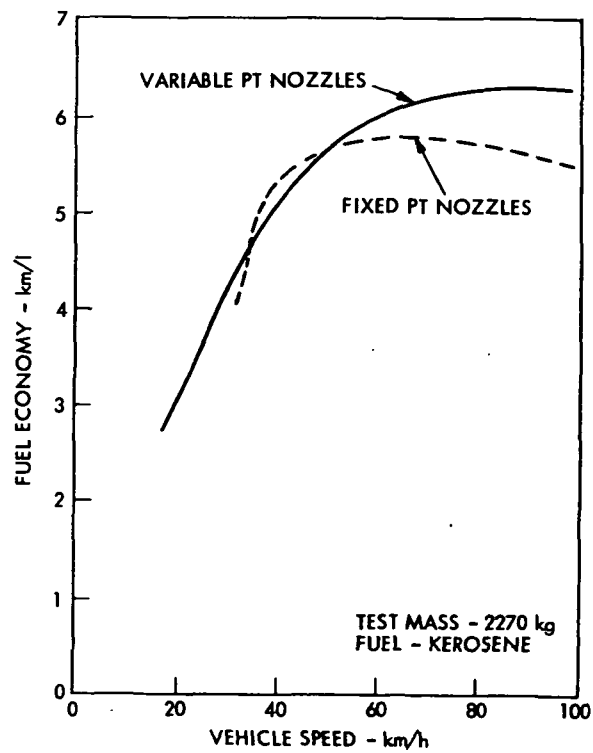


Figure 3-32. Effect of Variable-Power Turbine Nozzles on Vehicle Fuel Economy (Reference 3-19)

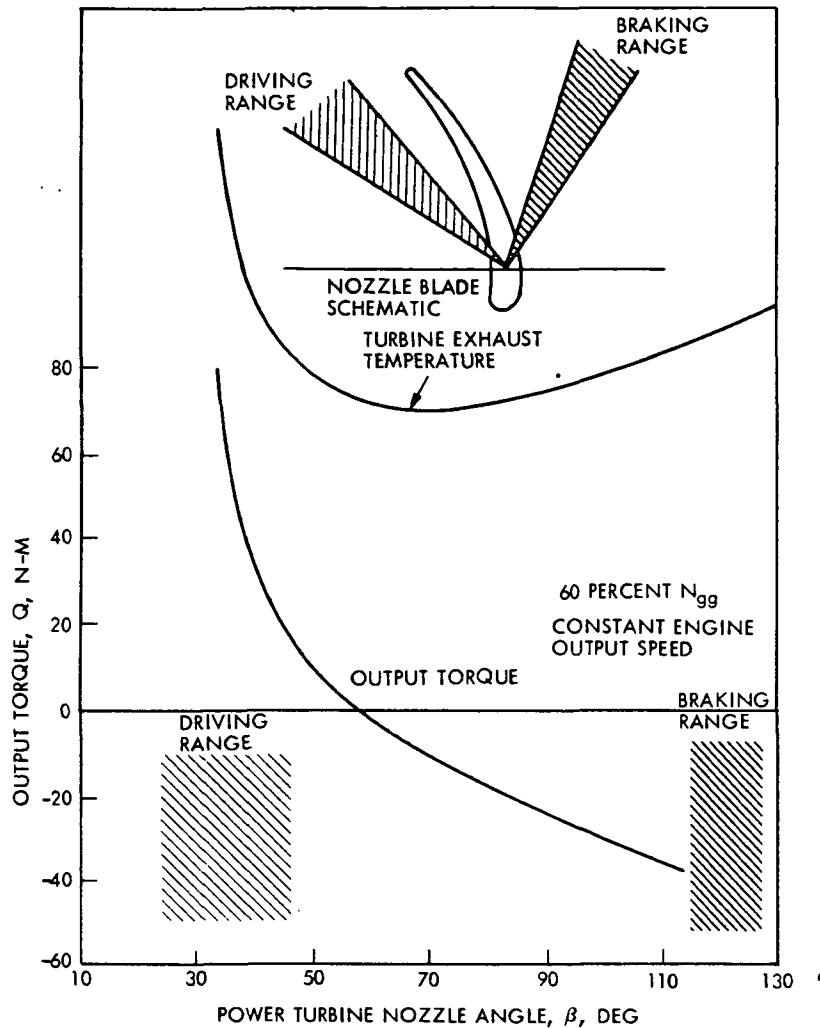


Figure 3-33. Effect of Nozzle Position on Output Torque and Heat Exchanger Inlet Temperature (Reference 3-60)

the trailing end of the vane is angled. The strong wakes produced by the flaps when angled and the close proximity to the compressor inlet suggest possible unfavorable effects on the compressor efficiency. In Configuration C the guide-vanes are located farther upstream of the compressor between the parallel walls of a radial inlet nozzle. This approach has the advantage that the vanes can be prismatic in shape, which requires less angular motion to produce the desired pre-swirl. The wakes produced by the vanes are less pronounced and can dissipate before they reach the compressor inlet. Detroit Diesel Allison has taken this approach in their AGT-2 single-shaft engine development program (Reference 3-26). Figure 3-36 shows approaches taken (A) and suggested (B) to implement a variable diffuser with the objective of flattening the characteristics of high performance radial compressors. The AGT-2 being developed by Detroit Diesel Allison is the only engine that implements concept (A) at present.

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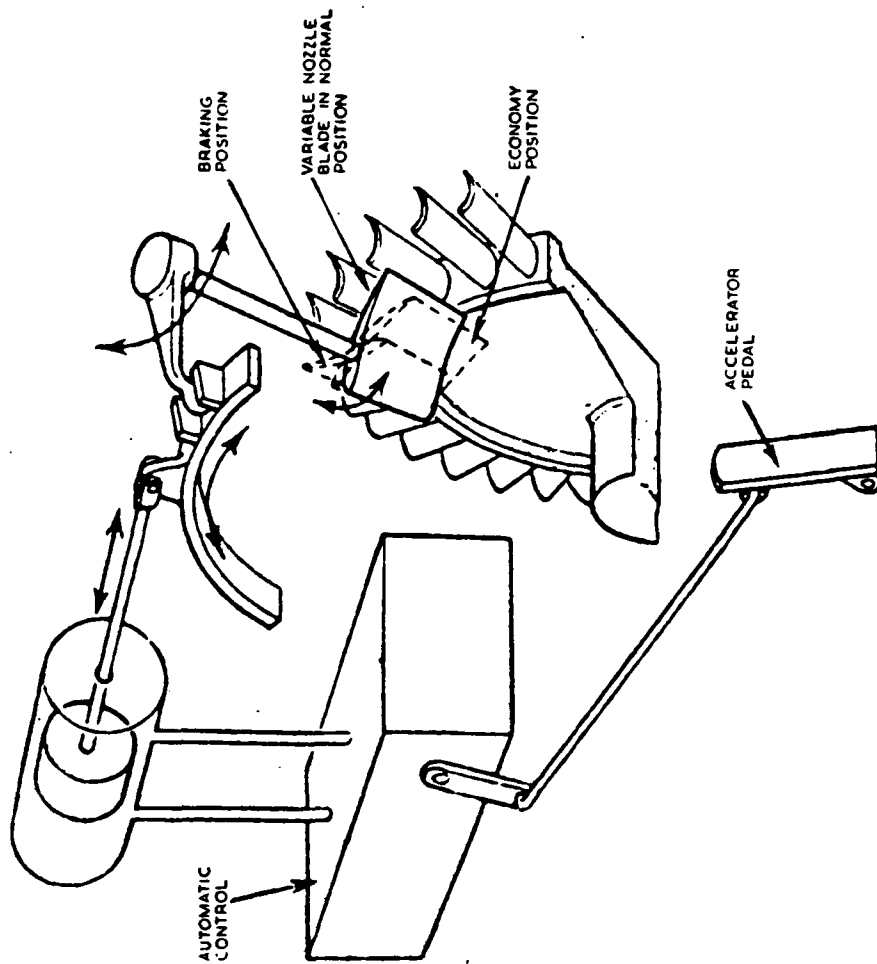


Figure 3-34. Schematic View of Chrysler Variable-Nozzle System (in part from The Gas Turbine Engine by Jan P. Norbye. Reprinted with permission of the publisher, Chilton Book Co., Radnor, PA. (Reference 3-58)

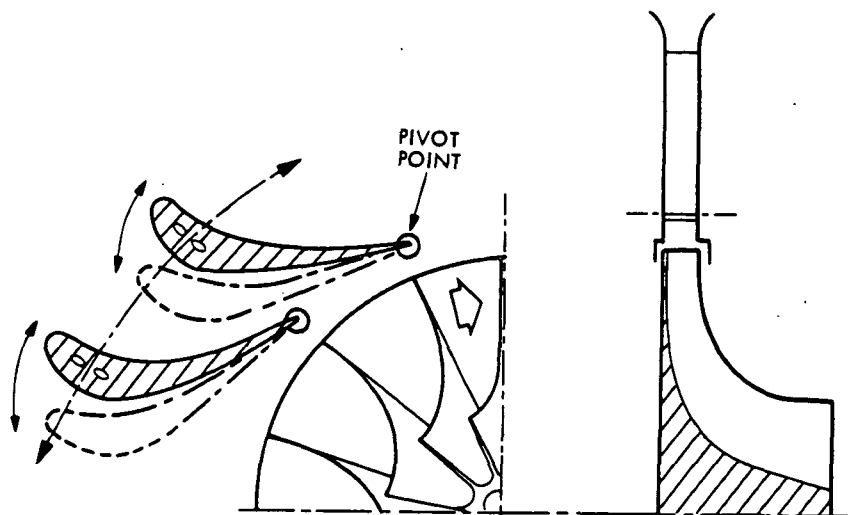


Figure 3-35. Typical Variable-Geometry Nozzle Arrangement for Radial Inflow Turbines

Variable-power turbine nozzles can considerably flatten the constant speed characteristics of two-shaft engines compared to those of fixed geometry, but this has not resulted in a significant improvement in vehicle fuel economy under variable load driving conditions. As will be discussed in more detail, to minimize the losses resulting from varying power demands, power changes should be accomplished in a manner that requires a minimum of speed, pressure, and temperature change. This can only be brought about by changing the flow-controlling throat area of the turbine inlet nozzle and the compressor diffuser inlet angle. Efforts in this direction have been under way for a number of years. According to a General Motors program (Reference 3-19) involving variable-nozzle turbines, the effectiveness of variable guide-vanes is limited in axial turbines, because it is difficult to maintain small radial clearances at sufficiently wide angular range. Also, at flow rates larger than nominal, the effects of nozzle angle and cross-sectional area changes decrease rapidly and thus the turbine rotor becomes the flow-controlling area.

The GM study (Reference 3-19) concludes that only radial inflow turbines have a potential for a truly wide range map-shift such as is necessary to achieve constant speed power-control. There are, however, a number of penalties that should be taken into account. The design of a nozzle can only be optimized for one given flow angle which usually corresponds to the operational conditions most frequently encountered. Clearance losses are relatively high (if compared to axial flow vanes) because of vane length and the need for wider tolerances. The tolerances depend upon controlling the distance between two parallel planes which are subject to bowing. This is primarily a problem of thermal deformation and redundant loading between the scroll housing, the vane pivot shoulders, and other associated hardware.

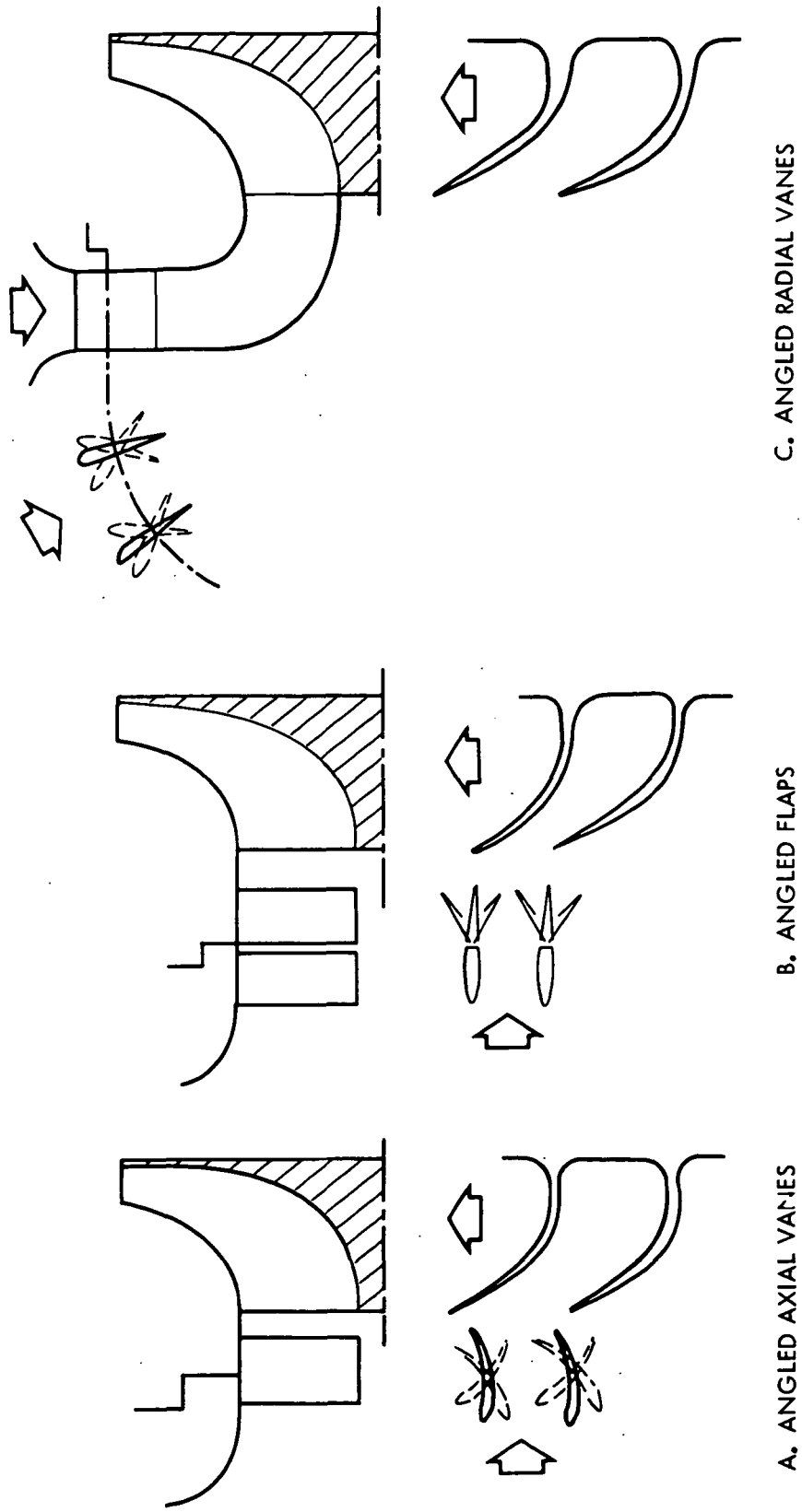


Figure 3-36. Typical Variable Guide-Vane Design Concepts for Compressor Inlet

Another problem is the choice of location for the vane pivot point. From a mechanical design standpoint it is desirable to pivot around a point where the vane is sufficiently thick to accommodate the pin. This, however, is associated with an undesirably large vaneless space between the rotor and the vane edge at small angles. A pivot point as close as possible to the rotor, on the other hand, leads to thick trailing edges. Either approach has losses associated with it. The minimization of these losses is one of the key design problems for variable nozzles. In most designs, the approach taken is to place the pivot point as close as feasible to the outer circle of the minimum vaneless space that is required for wake mixing and flow equalization (Figure 3-37).

Various ways of controlling the effective throat area of turbines by means of blocking (Figure 3-38) individual spaces between blades or entire portions of the nozzle circumference have been suggested and tried experimentally. Blocking techniques are attractive from the design standpoint because they are mechanically simple to implement compared to variable vanes. Blocking can be effective in cases where small changes of the total effective throat area can improve or remedy an undesirable condition such as overheating during starting transients, operations at part-loads, idling, etc. A disadvantage of blocking is the associated windage loss. It also introduces aerodynamic discontinuities and shock loads that are undesirable from the standpoints of dynamics, stress-fatigue, and service life. The application of the blocking technique is therefore limited in regard to the fraction of the total turbine inlet area that can be blocked and the total time the engine can remain in the blocked mode. Up to 25 percent blocking has been implemented experimentally on reaction turbines. Impulse turbines can be blocked up to a partial admission on the order of 2-3 blade spacings. Because of these circumstances, designers have shied away from the application of blocking techniques. However, for ceramic turbines and high temperatures, blocking might be the only practicable way of controlling the throat area.

3.3.6 Control Systems

Control of small fixed-geometry gas turbines can be achieved relatively simply by fuel metering using a hydromechanical governor. This controls the power output and assures operation within a safe envelope that would prevent turbine overheating, gasifier overspeed, and compressor stall (or surge). Mechanical centrifugal sensors and pressure taps provide all the necessary control information. Such systems are rugged and reliable and relatively inexpensive compared to the cost of the turbine.

Controlling today's variable-geometry automotive gas turbine is much more complicated. As is shown, for example, in Tables 3-2 and 3-3, and Figures 3-39 and 3-40, the requirements for controlling variable-geometry turbines are severe, and the systems necessary to accomplish this with a high degree of accuracy and reliability are more sophisticated and expensive than those needed in most other gas turbines

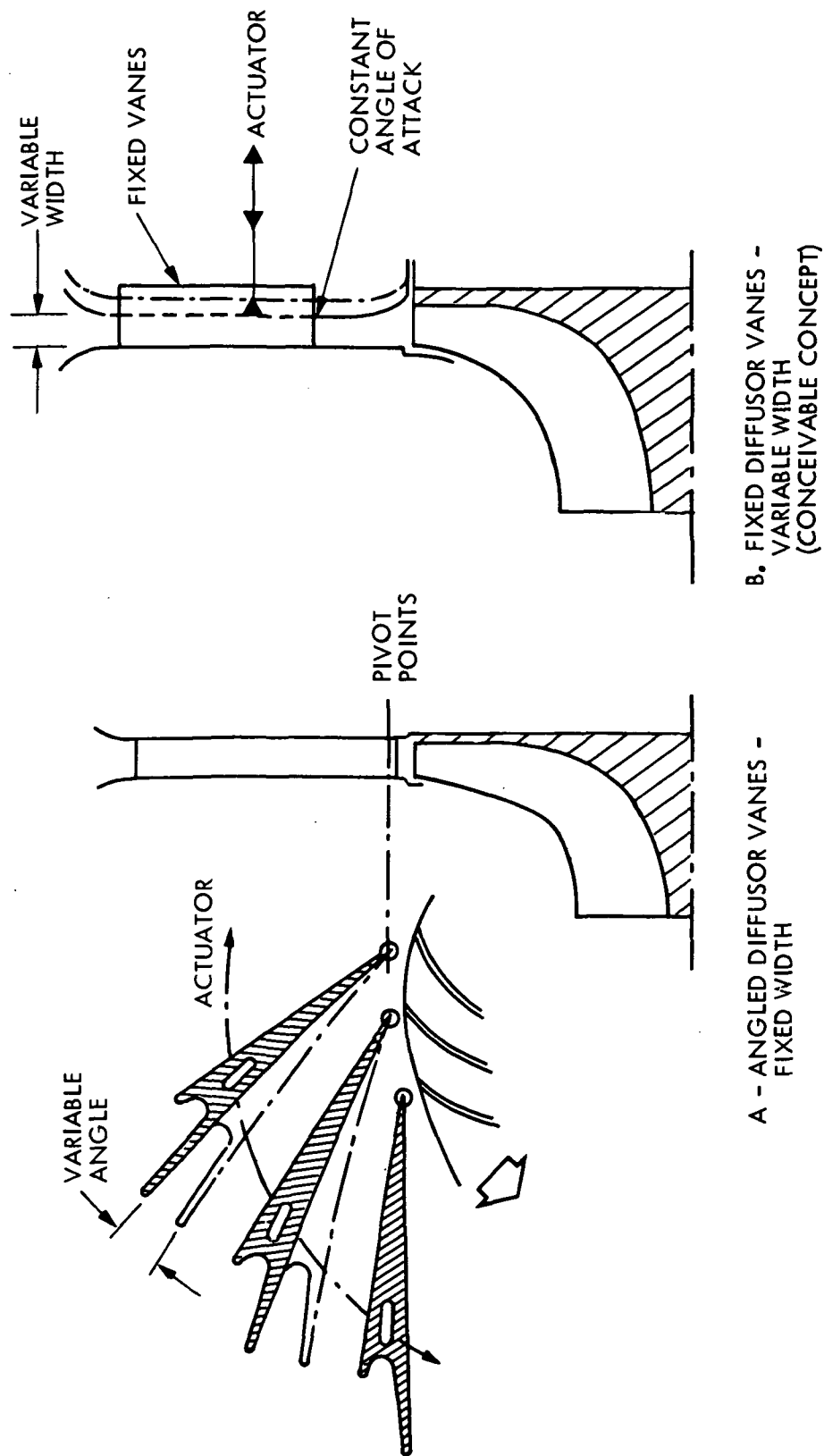


Figure 3-37. Typical Variable-Diffuser Design Concepts for Mach-Sensitive Radial Compressors

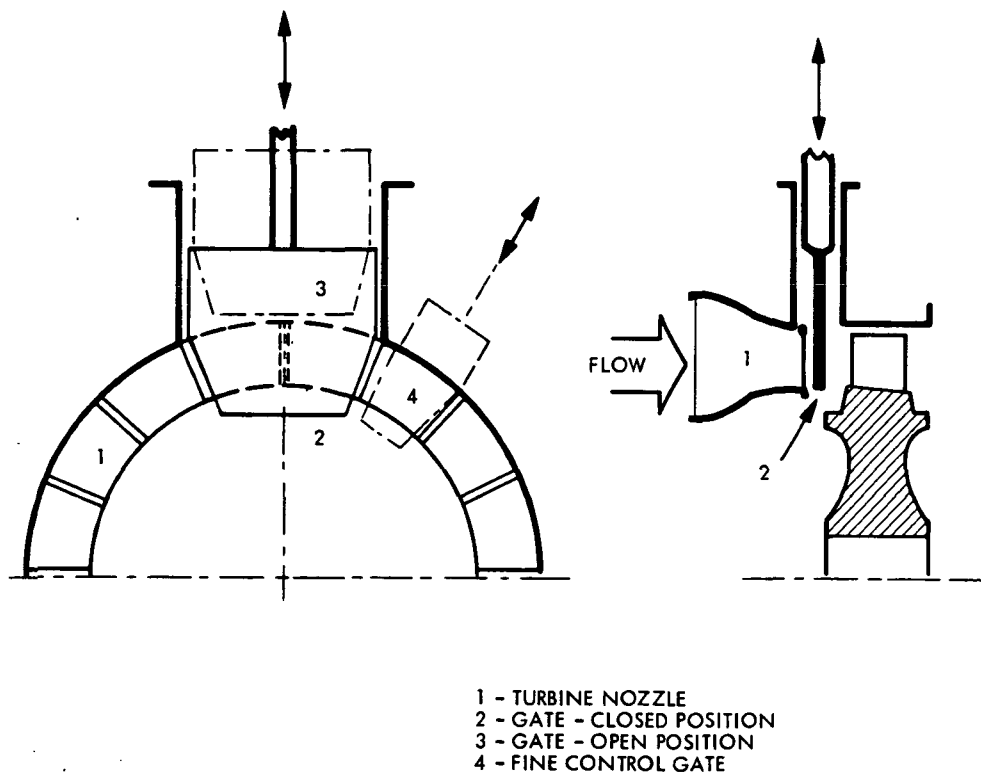


Figure 3-38. Nozzle Throat Area Reduction by Means of Blocking

applications. Technological advances (Figure 3-41) such as integrated circuits, miniaturized electronic components, and broad-based standardization should reduce the cost of engine control to a level that is acceptable for automotive gas turbine applications.

3.3.7 Gas Bearings

Gas bearings have been recently introduced into the gasifier of the Chrysler Upgraded engine on an experimental basis (Reference 3-27). The main advantages of gas bearings are lower parasitic losses, smaller orbital excursions of the shaft axis, improved hot soak capability, and longer service life. The development goal of the Chrysler-installed bearings, which are the self-starting type, is a service life of 3500 hours with 25,000 start/stop cycles; this corresponds to approximately 100,000 miles of vehicle operation. Component tests to study the effects of tolerances and clearances, various seal materials, and foil lubricants are in progress. For high temperature turbines, such as those using ceramic materials, air bearings will be a necessity from the hot soak standpoint and will have to be pressurized from some independent source, such as a battery-powered air pump, because the pressurization source will have to be run up before the engine can be started and cannot be shut off before the turbine has come to a full stop.

Table 3-2. Typical Automotive Gas-Turbine Control System Parameters (Reference 3-21)

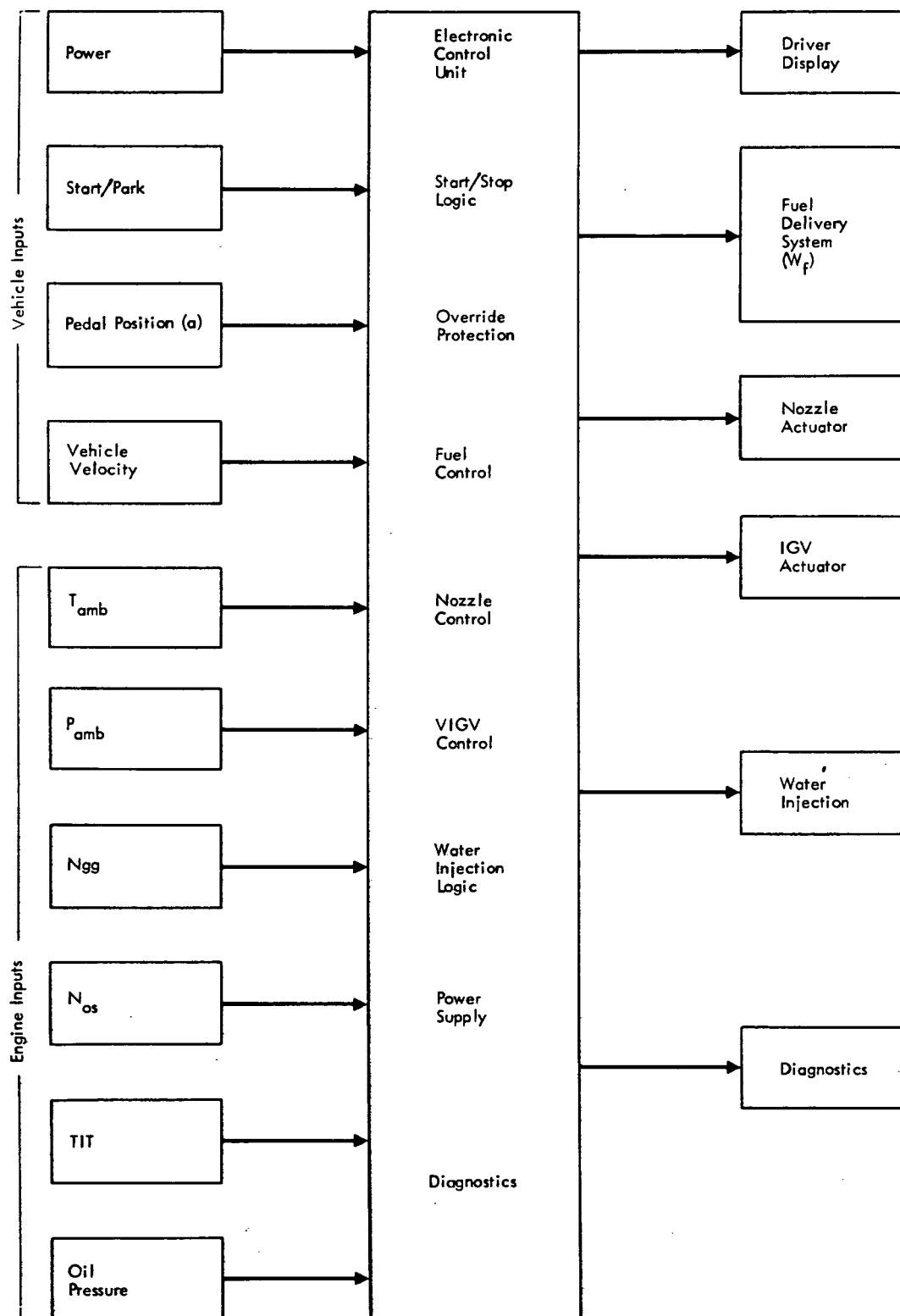
ENGINE AND CONTROL SYSTEM PARAMETERS

T_s	Gas generator turbine inlet temperature
T_{ex}	Turbine exhaust temperature
T_a	Ambient temperature
P_a	Ambient pressure
N_{gg}	Gas generator shaft speed
$\frac{N_{gg}}{\theta}$	Corrected gas generator speed
δ	Ambient pressure correction = $\frac{P_a \text{ psia}}{14.7 \text{ psia}}$
θ	Ambient temperature correction = $\frac{T \text{ }^\circ\text{R}}{545 \text{ }^\circ\text{R}}$
N_{os}	Output shaft speed
α	Throttle pedal position
β	Power turbine nozzle angle position
γ	Inlet guide-vane angle position
W_f	Fuel flow in pounds per hour
V_v	Vehicle velocity

3.3.8 Turbine Exhaust Diffusers

A component which can have a strong effect on engine fuel economy is the turbine outlet diffuser (Figure 3-42). Unfortunately, in automotive gas turbines, diffuser optimization conflicts seriously with engine size and weight. The highest diffuser efficiencies are obtained with plate diffusers (Configuration A) extending radially to a diameter of about 3 times that of the turbine. An optimized axial

Table 3-3. Typical Functional Diagram for Automotive Gas Turbine Control Systems (Reference 3-21)



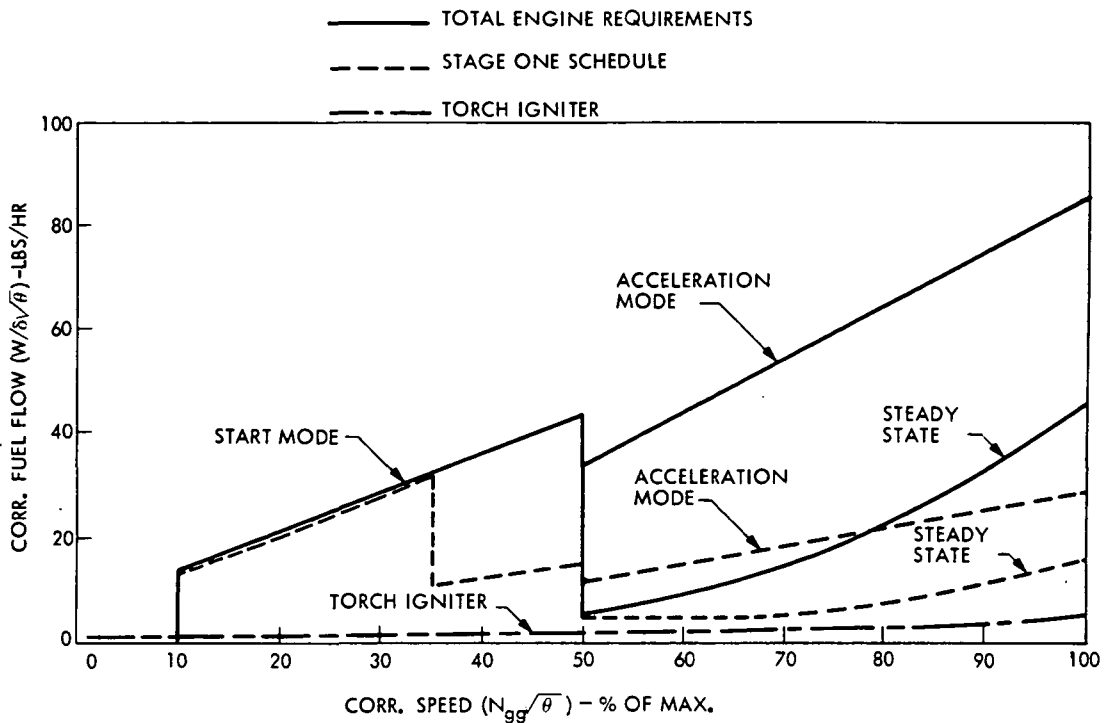


Figure 3-39. Typical Fuel Metering Schedules for Current Automotive Gas Turbines

diffuser (Configuration C) would have to extend downstream about 2-1/2 turbine outlet diameters and the power turbine shaft would have to extend through the center of the gasifier shaft out in front, as in the VW-GT-70 engine. Exponential diffusers (Configuration B) are efficient and space-effective, but dictate an in-line arrangement with a highly resistive system such as the heat exchanger. The approach that has emerged as the most practical for two-shaft engines is a rapidly diverging trombone-shaped diffuser (Configuration D) with guide-vanes in the axial/radial transition.

3.4 PRESENT STATUS OF DEVELOPMENT OF AUTOMOTIVE GAS TURBINE ENGINES

3.4.1 Introduction

In the previous sections, the advantages and disadvantages of the various gas turbine concepts and configurations were discussed and the characteristics and present status of the various components needed in the various gas turbine configurations were reviewed. In this section, the characteristics of complete engines and vehicles powered by state-of-the-art and advanced gas turbines are considered. To the extent possible, the results given in this section are based on actual test data from prototype engines, but in many instances it was

BASELINE ENGINE CONTROL SYSTEM

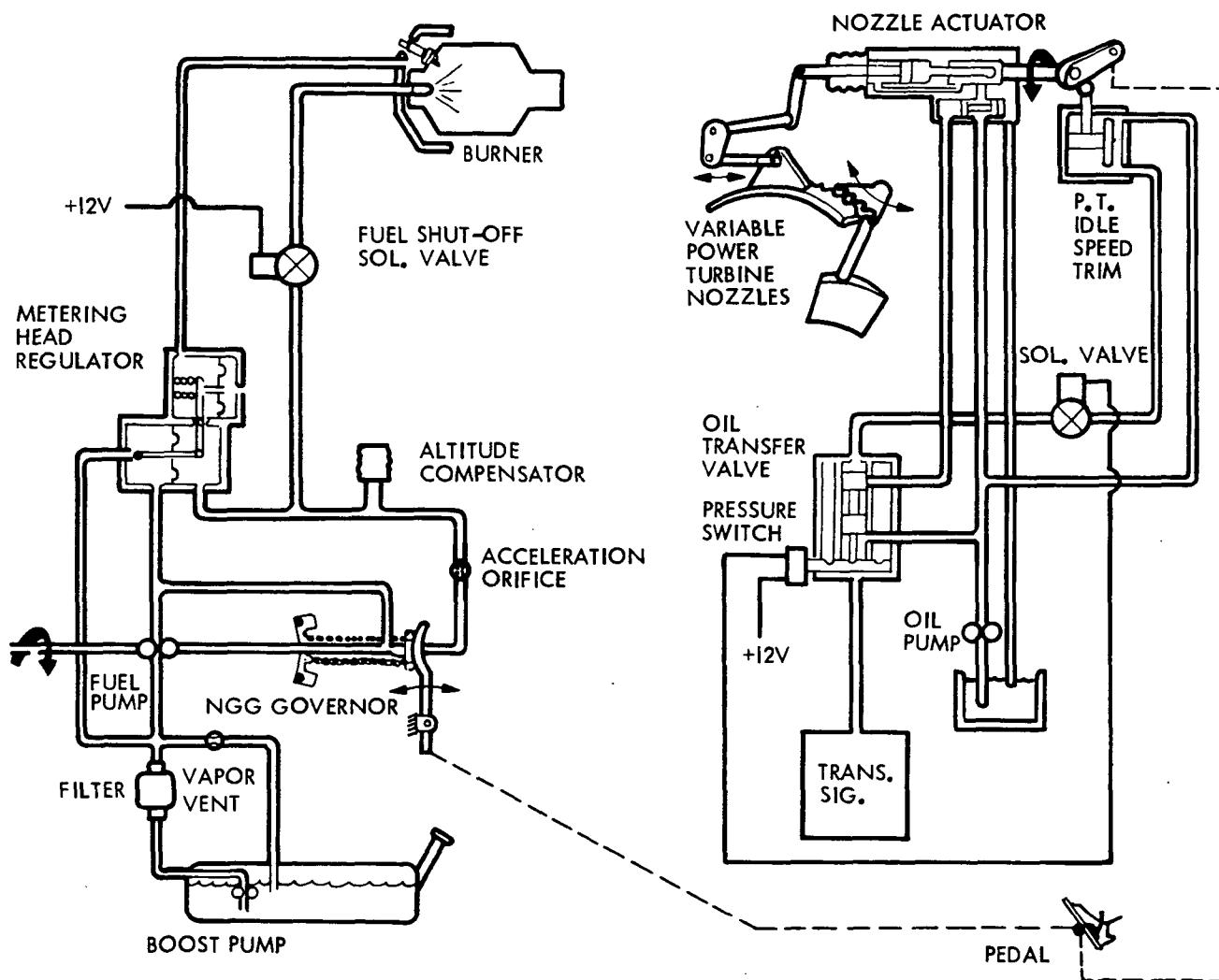


Figure 3-40. Schematic of Typical Variable-Nozzle Automotive Gas Turbine Control System (Reference 3-21)

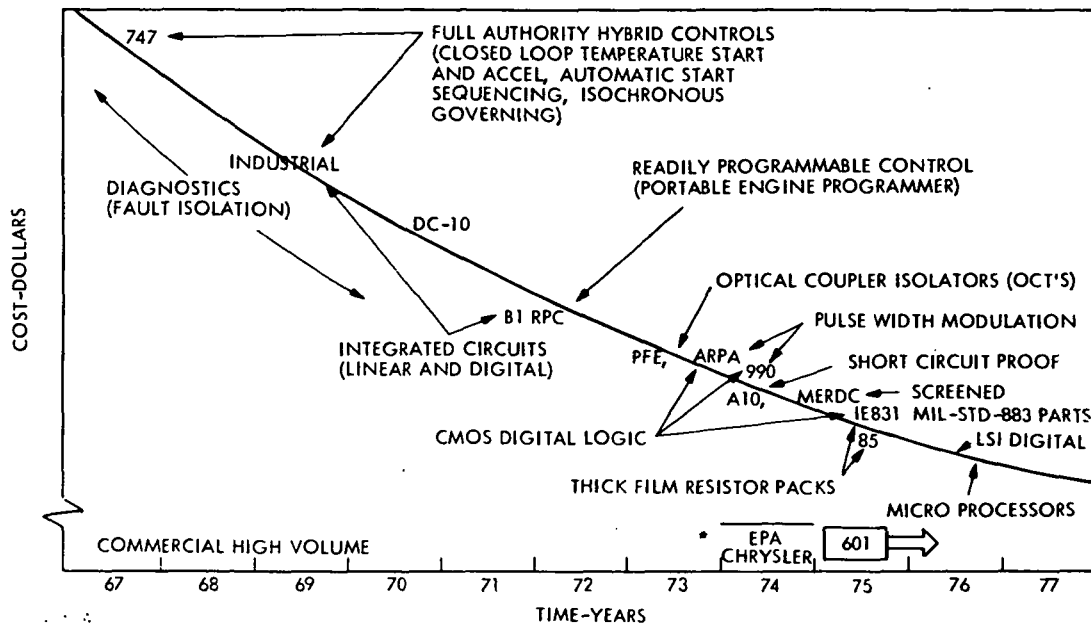


Figure 3-41. Effect of Electronic Innovations on the Cost of Control Systems

necessary to base projections on design calculations, extrapolations, and generalizations of the limited test data available and on computer simulations.

3.4.2 Engine Design and Development Programs

As indicated in Table 3-4, there are a number of gas turbine design and development programs currently under way in the world. Some of the programs are directed toward the development of engines for light-duty vehicles (passenger cars) while others are concerned with engines for long-haul trucks. The latter programs, involved with the development of 300-500 hp engines, have progressed through the engine and component design and testing phases and are now concerned with the road testing of prototype engines in long-haul truck service. The truck engine programs, which have been privately funded, have added to the information/data base for gas turbines intended for road use, but their direct relevance to gas turbines for use in passenger cars is quite limited, because of the large differences in size (50-100 hp vs. 300-500 hp) and duty cycle (stop-go driving at part load vs. near-steady operation at high loads) in the applications. However, as indicated in Tables 3-4 and 3-5, groups/companies which have in the past been involved primarily with the heavy-duty truck programs, are now also involved in both the design study and supporting R&D phases of the U.S. Department of Energy (DOE) program on automotive (light-duty) gas turbines. This involvement of the truck gas turbine groups brings to

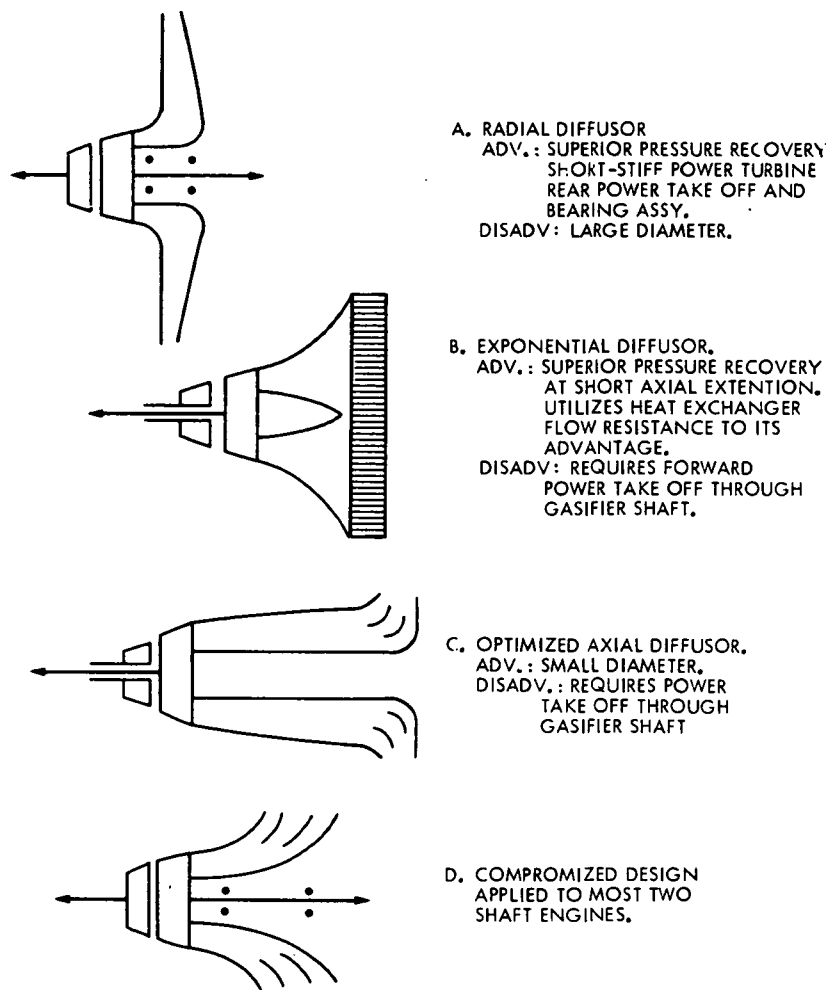


Figure 3-42. Typical Turbine Outlet Diffuser Concepts

the automotive program their extensive design and hardware experience. Likewise the recent involvement of the NASA Lewis Research Center in the DOE program brings to the program their extensive experience in aircraft gas turbines. Hence the current automotive gas turbine program is now benefiting from the combined expertise and experience of groups/companies with the broadest possible backgrounds and experiences in gas turbines.

Currently a number of gas turbine engines are being developed for use and/or have been tested in passenger cars. The overall characteristics of these engines, along with those of the two large truck engines being developed, are summarized in Table 3-6. All of the engines described in Table 3-6 utilize the two-shaft (free-turbine) configuration except for the Kronogard engine which utilizes a third shaft and a second power (or auxiliary) turbine. Single-shaft gas turbine engines for automotive use have been studied and preliminary "paper" designs developed (see Table 3-7) but none has as yet been

Table 3-4. Summary of Gas Turbine Engine Design and Development Programs

Developer	Engine Designation	Engine Type	Max. HP	Status	Country/Support
Chrysler	Upgraded	2 shaft, ceramic regenerator	104	Dynamometer tests	USA/DOE
	Baseline	2 shaft, metal regenerator	150	Vehicle tests	USA/DOE
VW/Williams	GT-70	2 shaft, ceramic regenerator	75	Vehicle tests	W. Germany/private
	GT-150	2 shaft, ceramic regenerator	140	Vehicle tests	W. Germany/private
United Turbine/Kronogard	KKT-MKI	3 shaft, ceramic regenerator	160	Dynamometer tests	Sweden/private
General Motors (DDA)	404/505	2 shaft, metal regenerator	300/400	Vehicle tests	USA/private
	AGT-2	Single shaft, non-regenerator, variable geometry	225	Component tests	USA/DOE
AiResearch	601	2 shaft, metal recuperator	500	Dynamometer tests	USA/private
Pontiac/DDA	-	-	-	Current study	USA/DOE
Williams/AMC	-	-	-	Current study	USA/DOE
AiResearch/Ford	-	-	-	Current study	USA/DOE

Table 3-5. Summary of Supporting R&D Activities for the
DOE Gas Turbine Program (Reference 3-25)

Contractor	Program Activity
General Motors (DDA)	Testing of variable geometry, single-shaft engine (AGT-2)
General Motors (DDA)	Incorporation of ceramic components into the DDA-404 engine
AiResearch	Durability testing of ceramic materials
Ford	Ceramic stator tests
Ford	Development of high density, reaction-sintered silicon-nitride material
Ford	Ceramic regenerator testing
Ford	Fabrication and testing of duo-density ceramic rotor
General Electric	Development of ceramic heat exchanger material
Orshansky	Development and testing of a hydro-mechanical CVT
Mechanical Technology, Inc.	Gas bearing development
NASA Lewis Research Center	Component testing in support of Upgraded Chrysler engine, computer simulation of single-shaft engines, test of fixed-ratio traction drive transmission, catalytic combustor tests

committed to hardware and tested in a passenger car, primarily because of the unavailability of a suitable continuously variable transmission. DOE has recently initiated a program with Detroit Diesel Allison (DDA) to obtain dynamometer test data for a single-shaft, variable-geometry engine of about 225 hp.

Most of the engine characterization and vehicle performance and fuel economy discussions in subsequent sections are based on information/data appropriate to the engines described in Tables 3-6 and 3-7 or advanced versions of those engines utilizing ceramic components.

Table 3-6. Characteristics of Gas Turbine Engines for Passenger Car and Trucks Applications

Developer	Chrysler	Chrysler	Volkswagen/Williams	Volkswagen/Williams	United Turbine/Kronogard	Detroit Diesel Allison	AIResearch Mack/KHD
<u>Engine Designation:</u>	Baseline (6th gen.)	Upgraded (7th gen.)	GT-70	GT-150	KTT-MKI	DDA-503	GT 601
Status	Vehicle tests	Component correction and dynamometer test tests	Vehicle tests	Vehicle tests	Component and engine dynamometer tests	Vehicle (truck) tests	Dynamometer tests
Max. HP	150	104	75	136	125	400	550
Max. Pressure Ratio	4.1	4.2	3.7	4.5	6	5	7
Max. Inlet Turbine	1850	1925	1675	1850	1900	1800	1800
Compressor type	Radial, VIGV	Radial, VIGV	Radial	Radial	Radial	Radial	Radial (2 stage)
Turbine type							
Gasifier	Axial	Axial	Axial	Axial	Axial	Axial	Radial
Power	Axial, VIN	Axial, VIN	Axial	Axial	Axial (both) VIN	Axial	Axial (2 stage), VIN
Combustor type	Single can	Pre mixed, torch ignition	Torodial	Single can	Single can	Single can	Single can
Heat exchanger							
Type	Dual rotary regen.	Single rotary regen.	Dual rotary regen.	Dual rotary regen.	Dual rotary regen.	Dual rotary regen.	Recuperator (4 modules)
Material	Metal	Ceramic	Ceramic	Ceramic	Ceramic	Metal	Metal
Controls	Mechanical	Electronic	-	Electronic	Electronic	Electronic	Electronic
Transmission type	3 spd. autom.	3 spd. autom.	3 spd. autom.	3 spd. autom.	2 spd manual electronic shifted	9 spd manual	Multi-ratio gear box
Minimum bsfc (^{lb} /bhp-hr)							
at max power	0.54	0.44 (design)	0.63	0.50	0.42 (calc.)	0.45	0.35 (estim.)
at 30% power	0.68	0.51 (design)	1.07	0.61	0.44 (calc.)	-	0.40 (estim.)
Weight (lb)	600	400	375	463	390 (est.) inc. transm.	-	2175
Dimensions (L x W x H), in.	36x28x30	33x33x27	24x26x16	9.7 ft ³	23x25x24	-	60x40x45

Table 3-7. Summary of Gas Turbine Engine Design Study Results (Reference 3-8)

Engine Type	2 Shaft, Regen. (Metal)	Single Shaft, Regen. (Metal)	Advanced Single Shaft, Regen. (Ceramic)
Max. HP	135	108	110
Max. pressure ratio	4.6	4.6	5.0
Max. inlet turbine temperature, °F	1850	1900	2400
Compressor type/ effic., %	Radial/81, VIGV	Radial/82, VIGV	Radial/86
Turbine type/ effic., %			
Gasifier	Radial/88	Radial/89	Radial/93
Power	Axial/88, VTN	-	-
Combustor type	Single can, fuel vaporizer	Single can, fuel vaporizer	Single can, fuel vaporizer
Heat exchanger			
Type/effic., %	Regenerator/90	Regenerator/90	Regenerator/94
Material	Ceramic	Ceramic	Ceramic
Transmission type	3 spd autom.	Traction- drive CVT	Traction-drive CVT
Min. bsfc (lb/bhp-h)			
at max. power	0.44 (calc)	0.41 (calc)	0.29 (calc)
at 10% power	0.55 (calc)	0.52 (calc)	0.33 (calc)
Idle fuel flow (lb/h)	3.4	2.7	2.3
Weight ⁽¹⁾ (lb)	470	380	300

(1) Includes battery, accessories, and exhaust system

3.4.3 Engine Characteristics

3.4.3.1 Size and Weight. The size and weight characteristics of various automotive gas turbine engines are presented in Table 3-8. Much of this information is the result of design studies (References 3-4, 3-5, 3-8, 3-28), since only limited hardware weight data is available in the open literature (Reference 3-29). As in most development programs, prototype gas turbine hardware has in general been heavier than the weight projected from design studies. For example, the Upgraded Chrysler engine was projected to weigh 400 lb based on the design study; however, the first experimental engine weighed approximately 600 lb (Table 3-9). Weight reduction studies have indicated

Table 3-8. Weight and Size Characteristics
of Automotive Gas Turbines

Developer	Engine Type	hp	Engine ⁽¹⁾ Weight	hp/lb	Engine Dimensions, in.			Source
					L	W	H	
Chrysler								
Baseline	2 shaft, metal	150	550-650	0.23-0.27	30	28	30	Ref. 3-21
Upgraded	2 shaft, metal	105	450	0.23	33	33	27	Ref. 3-5
VW/Williams								
GT-70	2 shaft, metal	75	375	0.20	24	26	16	Ref. 3-4
GT-150	2 shaft, metal	136	465	0.29	9.7 ft ³			Ref. 3-4
Kronogard								
MKI	2 shaft, metal	125	390 ⁽²⁾	0.32	23	25	24	Ref. 3-29
MKII	2 shaft, metal	85	280 ⁽²⁾	0.30	22	24	25	Ref. 3-29
Garrett 601	2 shaft, metal ⁽³⁾	550	2175	0.25	60	40	45	Ref. 3-30
Garrett Study								
	2 shaft, metal	135	470	0.29	-			Ref. 3-8
	Single shaft, metal	108	380	0.28	-			Ref. 3-8
	Single shaft, ceramic	108	300	0.36	-			Ref. 3-8

(1) Engine weight includes battery and radiator (plus water) when required.

(2) Included weight of a 2-speed electronically shifted gearbox.

(3) Recuperator; all others have regenerators.

(1) Engine weight includes battery and radiator (plus water) when required.

(2) Included weight of a 2-speed electronically shifted gearbox.

(3) Recuperator; all others have regenerators.

Table 3-9. Major Component Weight Breakdown for
the Chrysler Upgraded Gas Turbine as
Built (October 1976)

Component	Weight, lb
Engine housing	172
Interstage	26
Gas generator	57
Variable inlet guide-vanes	6
Air intake/starter clutch and motor	39
Burner cap and tube	24
Power turbine and reduction gear	154
Air and oil pumps	14
Regenerator cover	39
Regenerator core with gear and hub	32
Fasteners, lines, gaskets, filters, etc.	25
Power turbine nozzle actuator	6
Fuel control electronics	10
TOTAL ⁽¹⁾	604

(1) As of October 1976, Sixteenth Quarterly Progress Report,
Chrysler Corp.

that the weights of the housings, covers, reduction gear, etc. could be reduced to bring the engine weight closer to the projected value.

The weight characteristics of gas turbine engines of the free-turbine configuration are given in Figure 3-43, which shows engine weight as a function of horsepower. Measured and projected weight data are given for some current developmental gas turbine engines. The projected weight characteristic for fully developed gas turbine engines is represented by the solid line. This characteristic has been used in making the fuel economy projections for vehicles with gas turbine engines. Also shown in this figure are weights for the three-shaft engines being developed by Kronogard/United Turbine of Sweden. The weights for the Kronogard engines have been estimated by subtracting typical two-speed gearbox weights from the engine/transmission weight information available for these engines. Similar weight characteristics

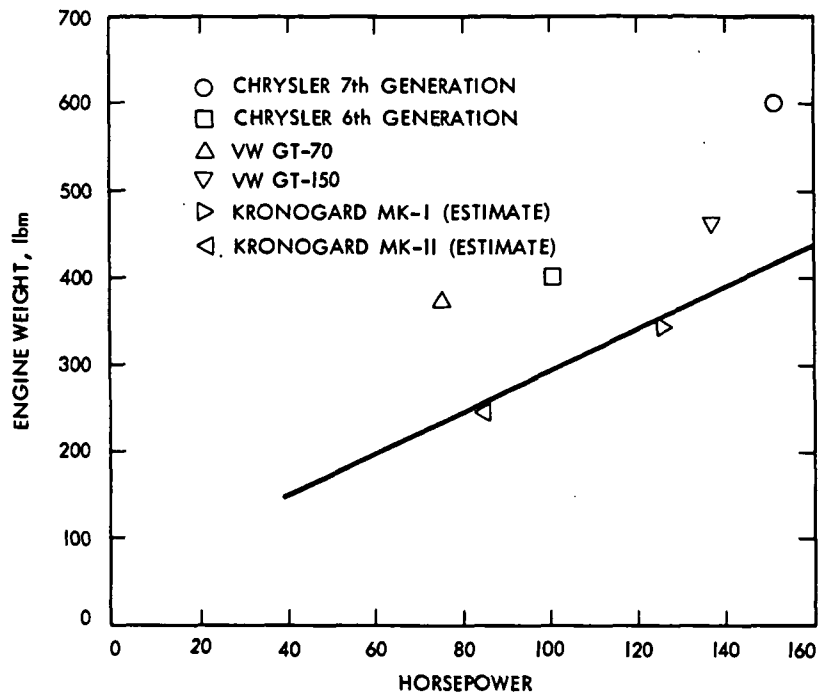


Figure 3-43. Engine Weights for Free-Turbine Brayton

for single-shaft gas turbines are given in Figure 3-44. Gas turbine engine weights are compared with those for other engine types in Section 9.

There is also considerable uncertainty relative to the size (volume) of gas turbines for use in automobiles. The power extraction elements (compressor and turbines) are relatively small, but the regenerator, combustor, and interstage ducts, and diffusers are rather bulky. Unfortunately, the overall efficiency of the engine depends critically on the flow in, and performance of those components, and it is not possible to minimize the volume of the engine and at the same time attain acceptable engine efficiency. Compromises in regenerator size and internal aerodynamics in order to reduce engine size can lead to large increases in system losses and greatly reduced efficiency.

3.4.3.2 Torque and Power Curves. The shape of the torque-speed characteristic of an engine is important in determining the acceleration capability of a vehicle. Torque curves for some of the (free-turbine) gas turbine engines under development are given in Figure 3-45. The engine torque and speed are normalized with respect to the torque and speed at maximum power. The torque curves are of special interest because they illustrate clearly the high torque of the (free-turbine) gas turbine at low engine speeds. The solid line represents the torque-speed characteristic which has been used in making the projections for gas turbine engines. This torque-speed characteristic is compared

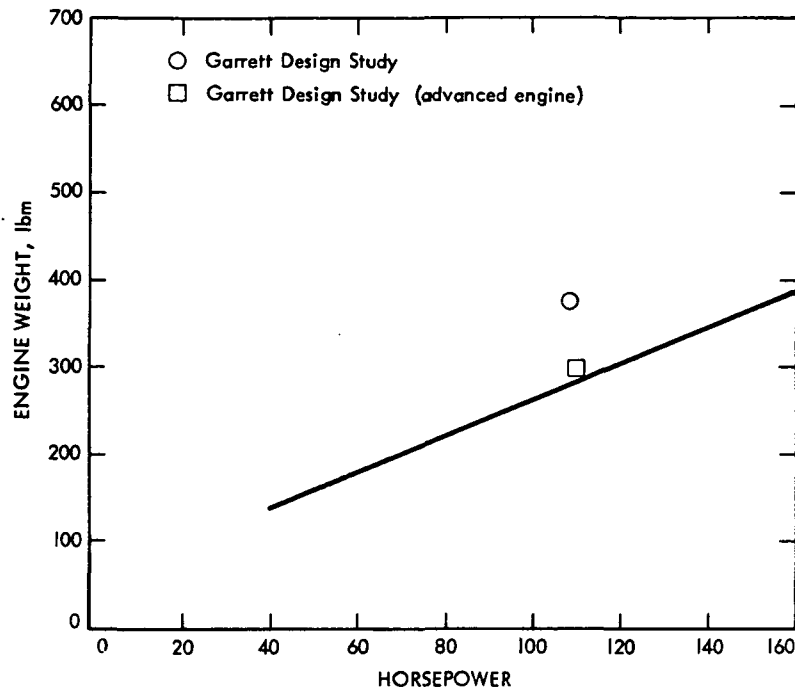


Figure 3-44. Engine Weights for Single-Shaft Brayton

with those of other engine types in Section 9. The corresponding power curves for some of the (free-turbine) gas turbine engines under development are given in Figure 3-46.

The torque-speed characteristic for the single-shaft gas turbine is shown in Figure 3-47. The torque curve for the single-shaft gas turbine is significantly different from that for the free-turbine configuration. The single-shaft gas turbine operates over a relatively narrow speed range and has low torque at low engine speed. Hence the single-shaft engine must be operated at high RPM regardless of vehicle speed and at near rated RPM whenever high power is required - for example, during acceleration and passing maneuvers. For these reasons the single-shaft engine must be used with a wide-range continuously variable transmission (CVT) in passenger car applications. The combination of a low inertia single-shaft gas turbine and a quick response CVT has attractive torque vs. vehicle speed characteristics and leads to the requirement for a vehicle power-to-weight ratio (based on 0-60 mph acceleration time) somewhat less than that needed with other engine types using standard transmissions.

3.4.3.3 Accessories. Unlike a conventional engine, the gas turbine does not require a radiator, and consequently does not suffer the parasitic losses associated with the water cooling system. Most of the waste heat from the gas turbine is rejected in the exhaust gases, which are relatively cool and at low pressure. The exhaust system is much

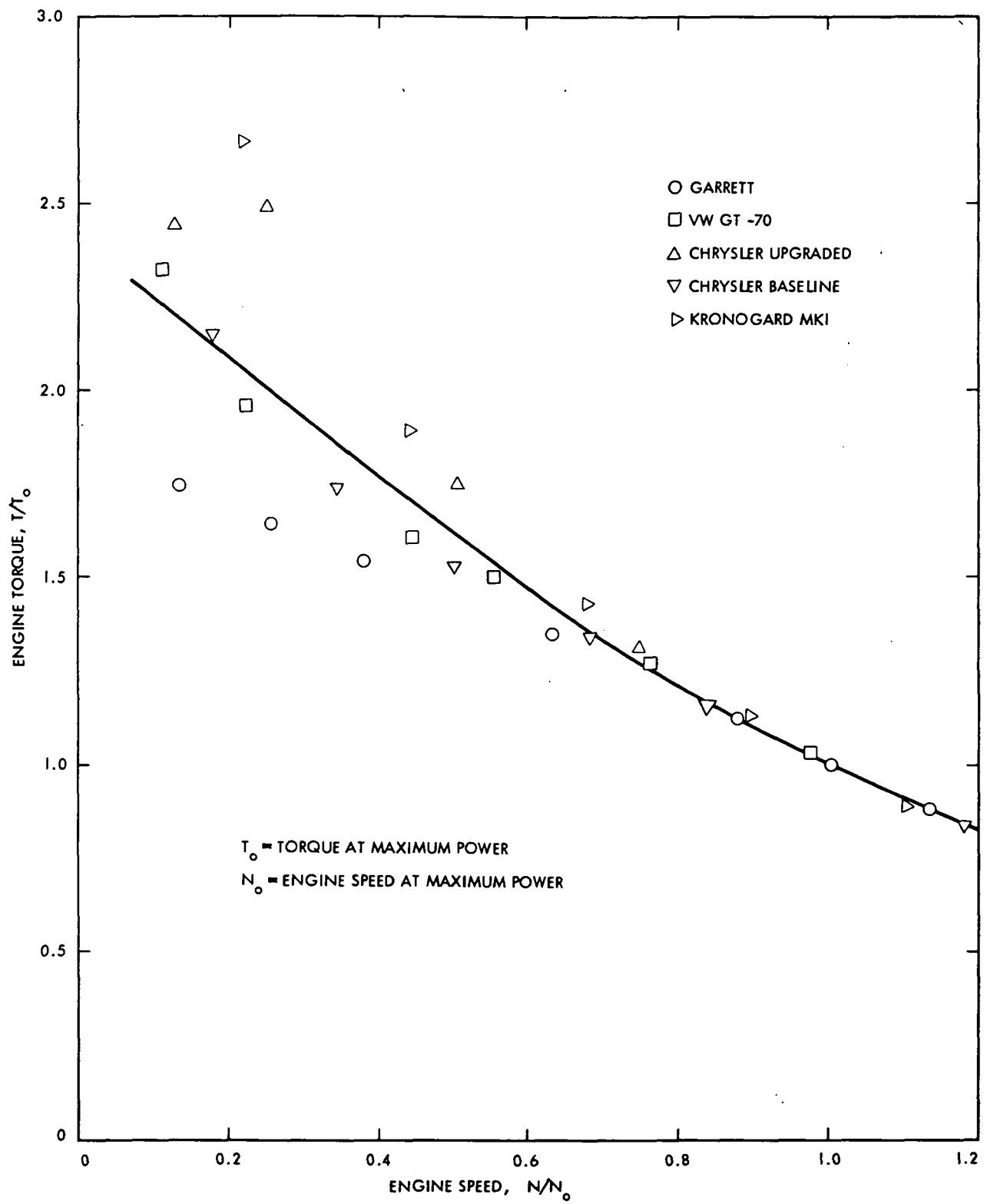


Figure 3-45. Torque-Speed Characteristics for Free-Turbine Brayton

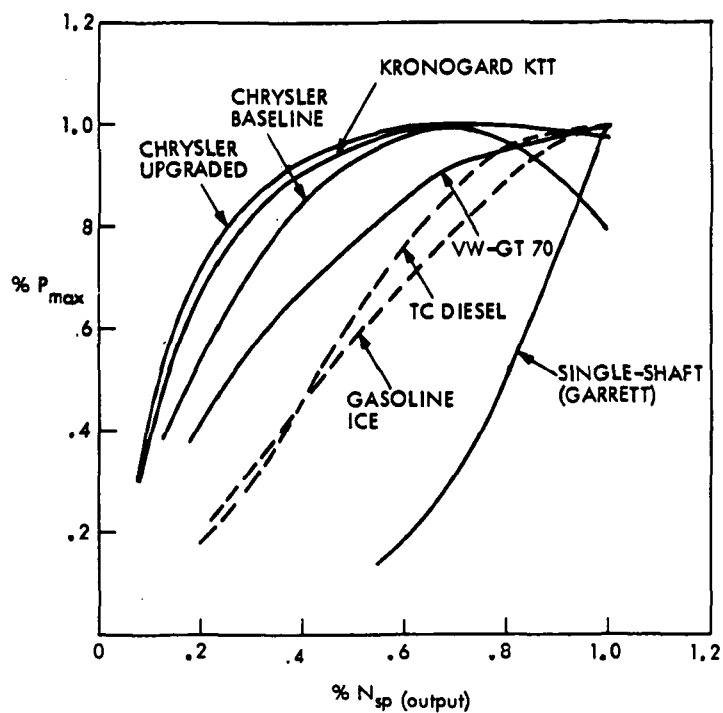


Figure 3-46. Power-Speed Characteristics of Various Gas Turbine Engines

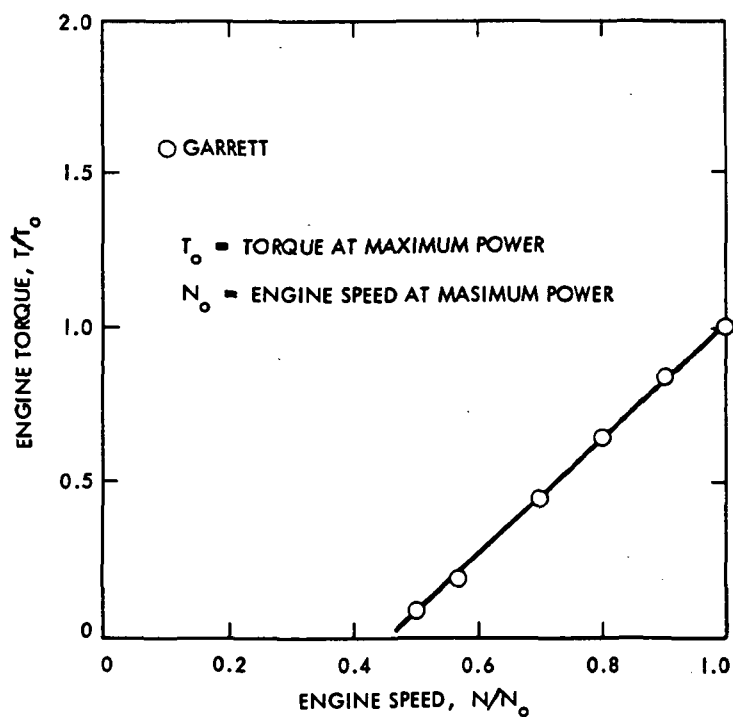


Figure 3-47. Torque-Speed Characteristics for Single-Shaft Brayton

larger than for an internal combustion engine, but no significant power losses result. Engine auxiliaries required by the gas turbine include the oil pump, fuel pump, and atomizing air pump. All of the engine auxiliaries are an integral part of the engine/cycle, and both analytical and test results must necessarily include their effect.

An important question in the design of two-shaft gas turbines is whether the accessories should be driven by the gasifier or power turbine shaft. Driving the accessories, including those required for passenger comfort and convenience, by the power turbine shaft results in a lower inertia gasifier and thus a more favorable engine response time, but requires a more complex accessory drive system because the power turbine operates over a wider RPM range. In the Baseline Chrysler engine, the accessories are driven by the gasifier whereas the Upgraded Chrysler engine utilizes a power turbine accessory drive system.

3.4.3.4 Engine Efficiency and Fuel Consumption Characteristics.

ENGINE EFFICIENCY

The effective efficiency of a gas turbine can be written as

$$\eta_{\text{eff}} = \frac{P_{\text{net}}}{E_F} = \eta_{\text{thermal cycle}} \eta_{\text{mech}}$$

where

P_{net} = net shaft power with all auxiliaries driven

E_F = fuel energy flow rate

$\eta_{\text{thermal cycle}}$ $\equiv$ cycle thermal efficiency

η_{mech} $\equiv$ mechanical efficiency of the gear system including the reduction gear

The relationship between η_{eff} and bsfc is

$$\text{bsfc}(\text{lb/bhp-h}) = 0.136/\eta_{\text{eff}}$$

for a fuel having an energy density of 18,600 Btu/lb. Calculation of η_{eff} for a regenerative single-shaft or multishaft gas turbine is not a simple matter, especially at part-load conditions, and requires the use of a rather complex computer program and detailed knowledge of component efficiencies for the complete range of engine airflow, turbine inlet temperature, and RPM. Results of such calculations for various proposed gas turbine designs can be found in the literature (References 3-7, 3-8, 3-9, 3-14, 3-32). Since detailed calculations of engine maps were not

undertaken at JPL as part of the present study, the efficiency (bsfc) maps published by the developers/proposers of the various engines were used in the vehicle/engine simulation studies discussed in later sections. These maps have not been independently verified.

Design-point engine efficiency calculations were done as part of the present gas turbine study. Such calculations are much simpler to make than those at part-load. As discussed in some detail in Section 3.8, a computer program was developed to treat the effects of engine size, mass flow, and turbine inlet temperature on losses and component efficiencies. That computer program was used to assess the effect of engine horsepower and increased turbine inlet temperature (i.e., the introduction of ceramic components) on gas turbine cycle efficiency.

In designing a regenerative gas turbine, the primary factors affecting engine efficiency are as follows:

- (1) Maximum pressure ratio.
- (2) Compressor efficiency.
- (3) Turbine efficiency.
- (4) Turbine inlet temperature.
- (5) Regenerator heat recovery effectiveness.
- (6) Cycle leakage.

The effect on engine cycle efficiency of varying each of the above parameters (one at a time from a nominal value) is shown in Figure 3-48. It is clear that each of the parameters individually can have a significant effect on cycle efficiency. In designing gas turbine engines, considerable care must be taken not to improve one parameter at the expense of degradation in one or more of the others.

As discussed in Section 3.8, the question of the effect of, and the interaction between engine design horsepower and turbine inlet temperature on gas turbine cycle efficiency is not easily answered for engine horsepower levels between 40 and 120 hp and turbine inlet temperatures between 1900 and 2500°F. To a significant extent, the attractiveness of gas turbines in passenger cars hinges on the answer to this question as the downsized vehicles of the 1980s will require horsepower levels in that range, and an increase in turbine inlet temperature up to about 2500°F using ceramics is thought to be required to meet fuel economy goals. A relatively complete introductory discussion of the effects of engine size/component finishing/turbine temperature interactions on component and engine efficiencies is given in Section 3.8. In addition, the approach used to include the complex interactions in the efficiency calculations is also presented. In this section only the principal results of the analysis and their significance in assessing gas turbines for use in passenger cars are discussed.

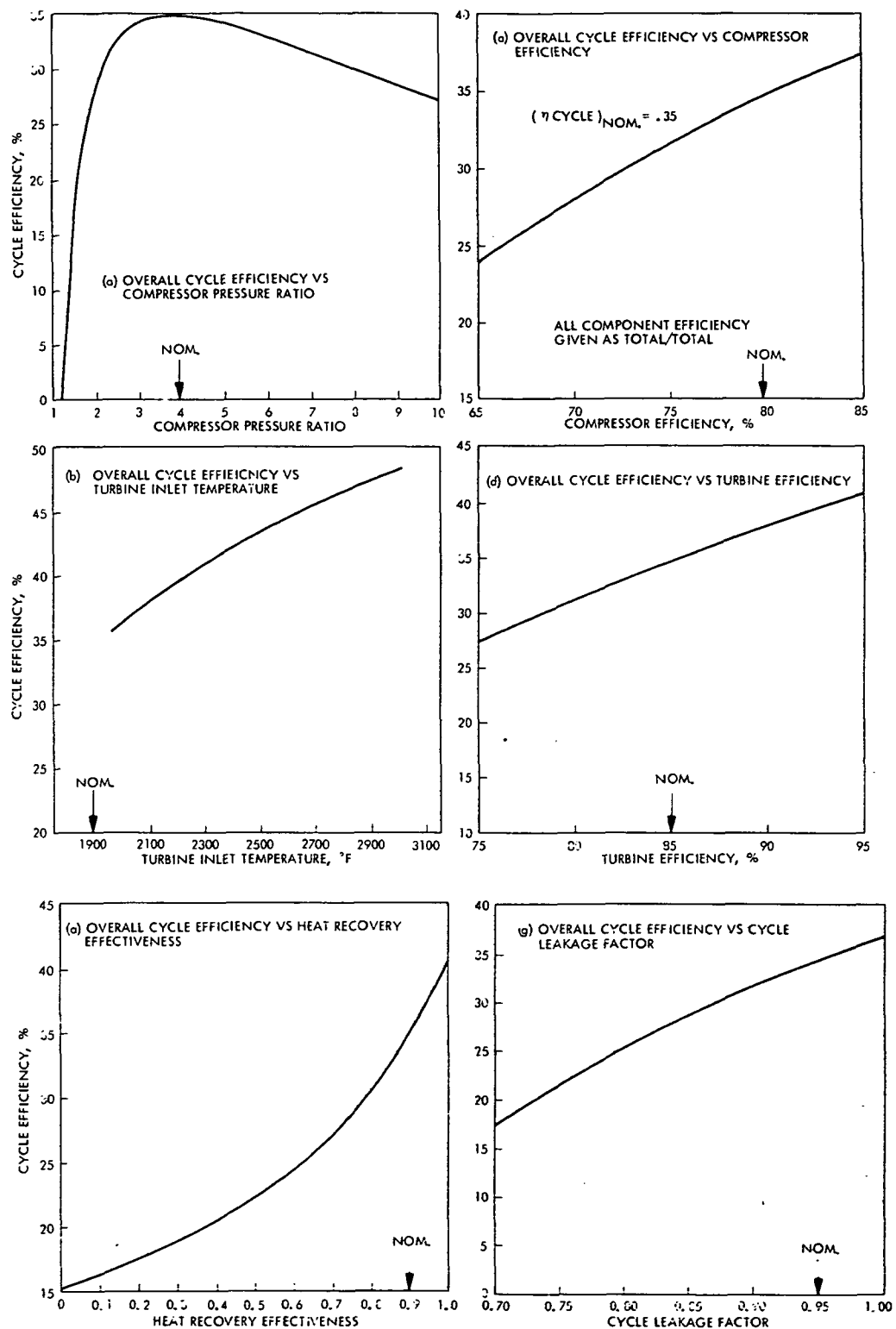


Figure 3-48. Effect of Parameter Variations on Engine Cycle Efficiency (Reference 3-15)

In the present analysis of size effects, the Chrysler Upgraded engine (100 hp and 1925°F turbine inlet temperature) was taken as the reference engine, and changes in component efficiencies were calculated relative to the design goals of that engine (Table 3-10). In essence, it is assumed that the 100-hp metal engine is just size critical and that further reductions in engine size would result in size-related degradations in component efficiencies.

Table 3-10. Design Parameters of 100-hp Engine Based Upon Reference 3-5

Item	Value	Dimensions
Compression ratio	4.0	--
Turbine inlet temperature	1925	°F
Power concentration (nominals unaugmented)	77.6	hp-s/lb
Specific fuel consumption	0.440	lb/hp-h
Efficiencies:		
Compressor	0.785	--
Combustor	0.990	--
Turbine	0.840	--
Heat recovery	0.900	--
Mechanical	0.900	--
Pressure drop:		
Inlet	0.015	--
Compressor to turbine	0.024	--
Turbine to atmosphere	0.090	--
Heat Losses:		
Regenerator to combustor	0.680	Btu/lb
Combustor to turbine	0.110	Btu/lb
Turbine to regenerator	0.230	Btu/lb
Regenerator leakage	0.045	--

The effect of turbine inlet temperature and design power output on engine flow rate relative to the reference engine is shown in Figure 3-49. The reduction in flow rate results in designs having smaller components with the associated lower Reynolds number, hydraulically rougher surfaces, increased fractional clearances, increased trailing edge thicknesses, and surface contour deviations. The calculated effects of these size-related factors on compressor, turbine, and regenerator leakage losses are given in Figure 3-50 as a function of engine power and turbine inlet temperature. The points shown as $\diamond$ correspond to the design goals for the Chrysler Upgraded engine. The calculated specific fuel consumption relative to the 100 hp metal reference engine is shown in Figure 3-51. Also shown in the figure is the improvement in bsfc which would result from increasing turbine inlet temperature (TIT) if size effects were neglected. The predicted improvement in bsfc for that idealized case is about 19 percent for increasing TIT from 1900°F to 2500°F. Including size-effects, the predicted improvement in bsfc from using ceramics (TIT = 2500°F) is reduced to 13.5 percent. It is also of interest to note from Figure 3-51 that a ceramic 50-hp gas turbine is predicted to have about the same bsfc as the Chrysler Upgraded engine. In other words, the component degradations due to size effects in reducing horsepower from 100 to 50 almost exactly offset the improvement due to increasing the turbine inlet temperature from 1900°F to 2500°F.

The results in Figure 3-51 were used in the vehicle simulation studies of fuel economy to correct the gas turbine engine fuel consumption maps for the effects of engine size and turbine inlet temperature. The correction factors were applied uniformly to the engine maps.

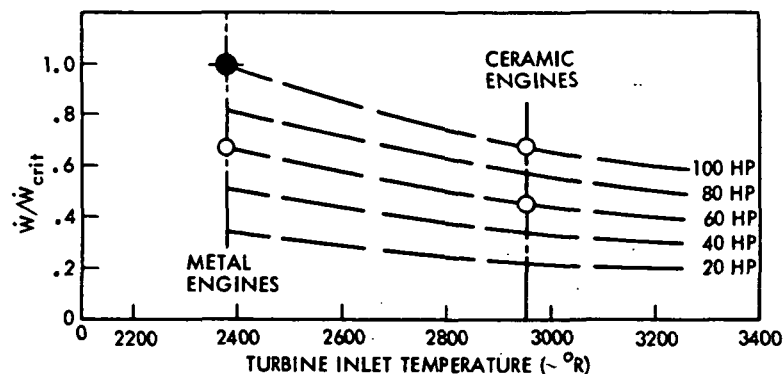


Figure 3-49. Effect of Turbine Inlet Temperature and Power Output on Engine Flow Rate Relative to Reference Engine (100 HP/1925°F)

$\diamond$ Reference Engine
 O Discussed Data Points

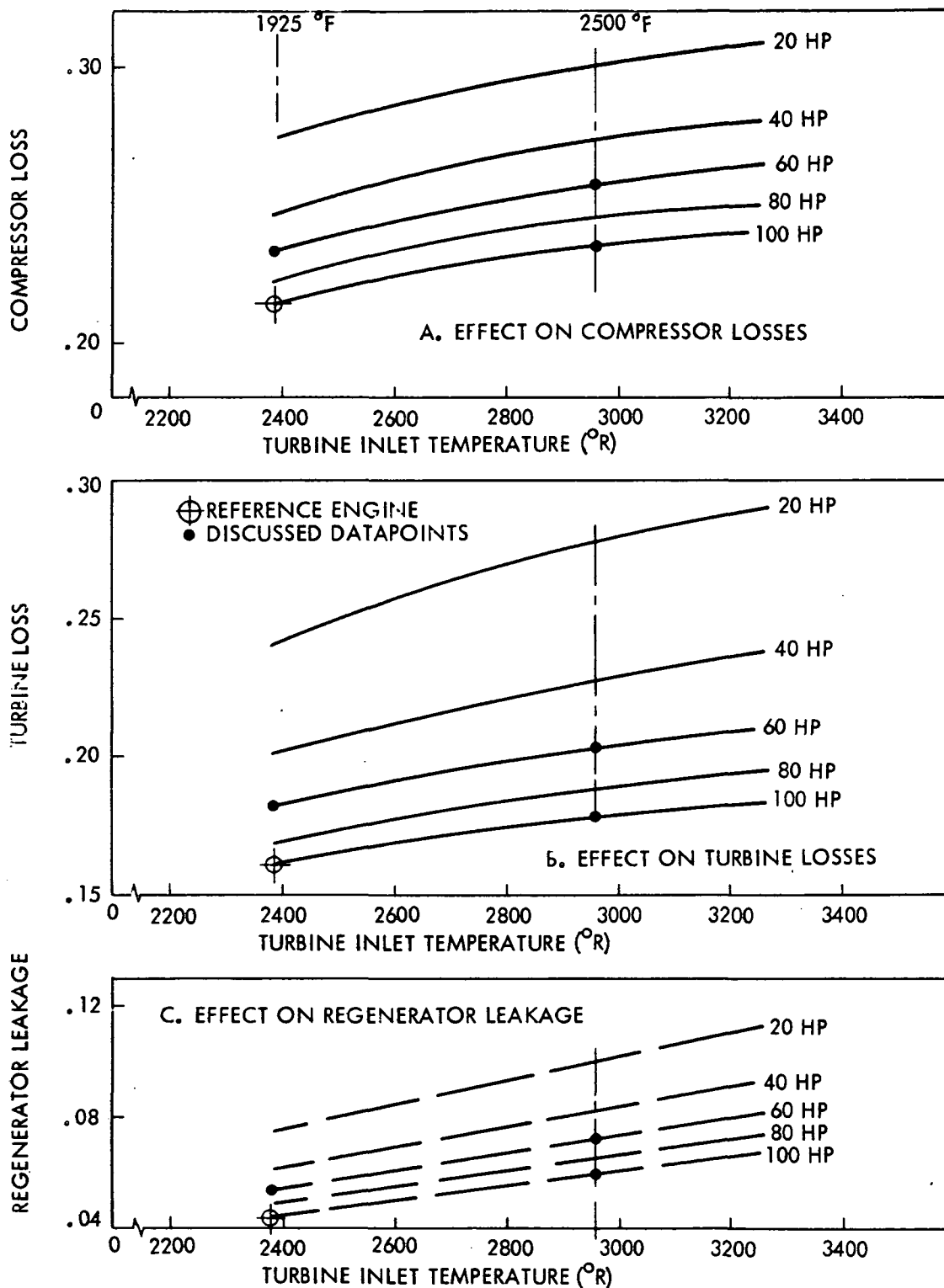


Figure 3-50. Effect of Turbine Inlet Temperature and Power on Change of Engine Losses

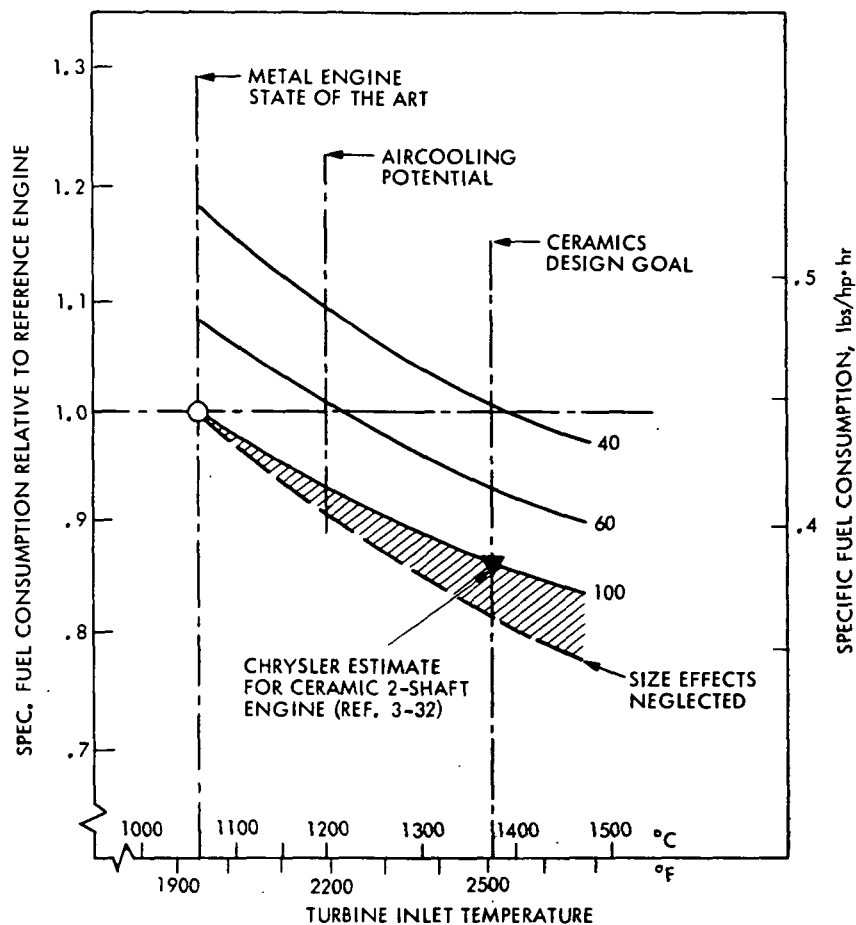


Figure 3-51. Effect of Turbine Inlet Temperature and Design Power on Engine Specific Fuel Consumption

FUEL CONSUMPTION

Information/data on the fuel consumption characteristics (bsfc) of gas turbine engines is rather limited. In part this is due to the dependence of the bsfc map on the choice of operating strategy (e.g., gasifier RPM, turbine inlet temperature, variable-geometry settings of variable inlet guide-vanes and variable turbine nozzles).

Bsfc maps for a complete range of output power and RPM are available in the literature for several gas turbine engines (e.g., VW GT-70, Kronogard KTT, and Garrett "study" engines). Several typical published maps are shown in Figures 3-52 to 3-54. For the most part, the bsfc maps available are primarily based on calculations and not extensive dynamometer testing of prototype engines.

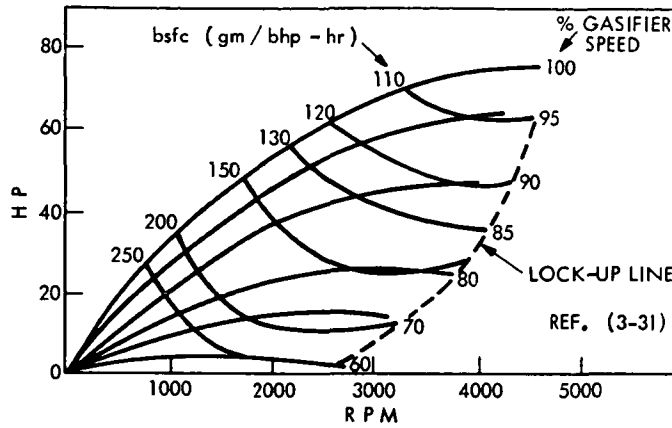


Figure 3-52. Fuel Consumption Characteristics of the VW GT-70 Gas Turbine

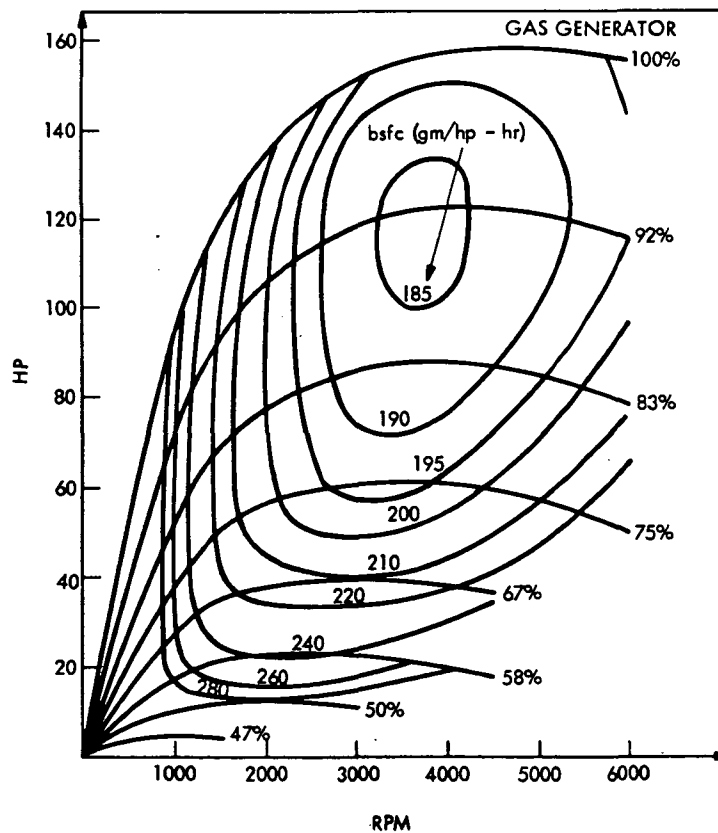


Figure 3-53. Fuel Consumption Characteristics of the Kronogard KTT Gas Turbine

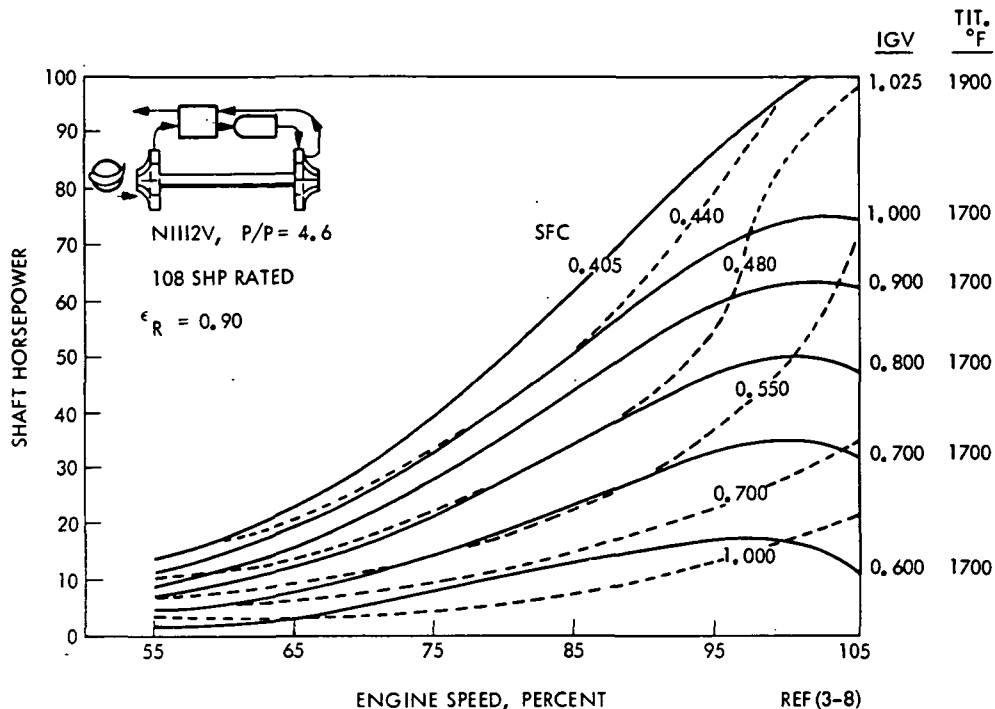


Figure 3-54. Fuel Consumption Characteristics of the Garrett Single-Shaft Gas Turbine

Fuel consumption maps for the Chrysler engines are not available. The steady-state bsfc map shown in Figure 3-55 was constructed for the Chrysler Upgraded engine using available information on the minimum bsfc and limit power curves and following the general map characteristics exhibited by the typical maps of Figures 3-52 to 3-54. This same procedure was followed to obtain bsfc maps for other engines for which only minimum bsfc and limit power curve information was available.

Tables of bsfc as a function of output RPM fraction ($\text{RPM}/\text{RPM}_{\text{max}}$) and output power fraction ($\text{HP}/\text{HP}_{\text{max}}$ at RPM) were prepared for a number of gas turbine engines for use in the vehicle/engine fuel economy studies. Bsfc information is given in Tables 3-11 to 3-16 for the following engines: Chrysler Baseline, Chrysler Upgraded (metal), Chrysler Upgraded (ceramic), Kronogard KTT, Garrett single-shaft (metal), and Garrett single-shaft (ceramic).

Information concerning engine idle fuel flow and rotational inertia is also needed in the vehicle/engine simulation studies. High idle fuel flow and effective rotational inertia (response time) can be sources of difficulty for using gas turbines in passenger cars. Most of the gas turbines tested to date in cars have had high idle fuel flow and high rotational inertia, but projections of those quantities for future automotive gas turbines indicate much lower values. Typical idle fuel flow and rotational inertia values for automotive gas turbine engines are given in Table 3-17. Idle fuel flow rates are shown plotted versus horsepower in Figure 3-56. The solid curves represent the idle flow

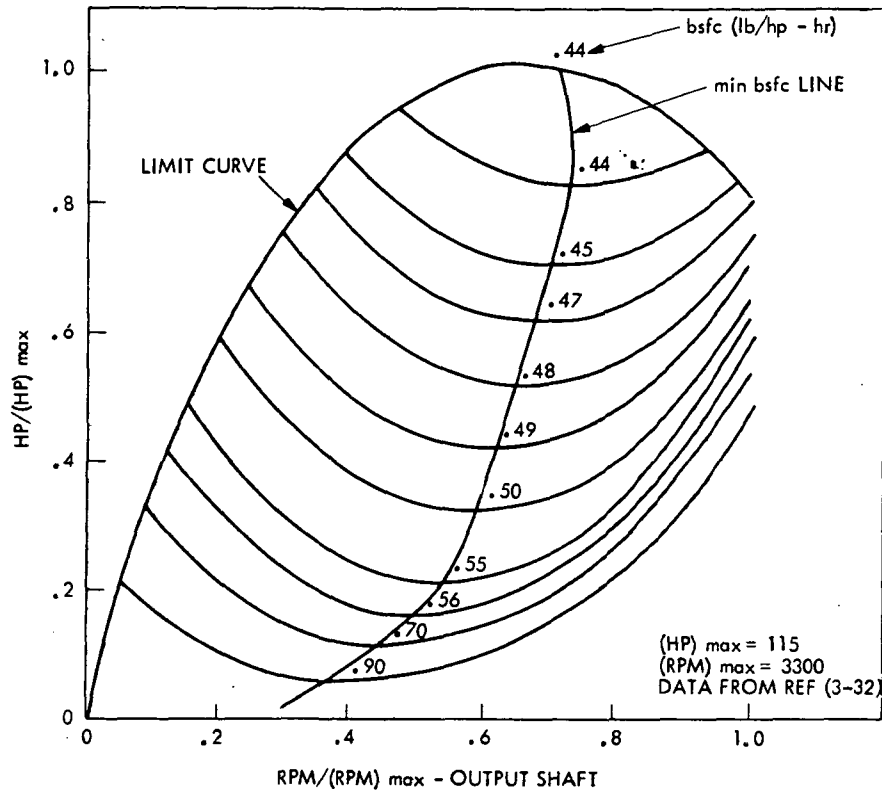


Figure 3-55. Fuel Consumption Characteristics for the Chrysler Upgraded Gas Turbine

rate characteristics which have been used to represent the gas turbine engine in this study.

3.4.3.5 Emissions. The emissions from a gas turbine engine are highly dependent on combustor design and fuel flow control strategy during the various engine operating modes. Experience to date has indicated that sufficiently low HC and CO emissions can be achieved to meet the vehicle standards of 0.4 g/mi HC and 3.4 g/mi CO using a single-can combustor with a spray fuel injector and secondary air dilution to obtain the desired turbine inlet temperature. Unfortunately, the NO_x emissions from the simple single-can combustor (droplet, diffusion flame combustion) are too high to meet the 1.0 g/mi NO_x standard. There have been numerous studies that show that gas turbine combustors can be developed which have simultaneously low HC, CO, NO_x emissions permitting a gas turbine-powered vehicle to meet the 0.4 gm/mi HC, 3.4 gm/mi CO, and 0.4 gm/mi NO_x standards. The low emissions combustor requires lean combustion utilizing a premixed, prevaporized fuel and a maximum temperature below that at which NO_x begins to form in significant concentrations. Such a combustor (Figure 3-22C) has been developed for the Chrysler Upgraded engine. Detailed emissions data for that engine are not yet available, but data (Reference 3-33) taken with the prototype combustor

Table 3-11. Fuel Consumption and Power Limit Curve Data
for the Chrysler Base Gas Turbine

BRAKE SPECIFIC FUEL CONSUMPTION MAP DATA (lb/bhp-h)*										
f_{RPM}	f_{HP}									
	0.10	0.15	0.25	0.40	0.60	0.80	1.00			
0.12	4.000	3.000	2.000	1.600	1.300	1.000	0.948			
0.20	2.500	1.650	1.330	1.000	0.887	0.800	0.730			
0.30	1.400	0.2000	0.958	0.833	0.730	0.676	0.630			
0.40	1.200	0.958	0.830	0.725	0.665	0.615	0.593			
0.50	1.000	0.915	0.790	0.670	0.610	0.592	0.560			
0.60	1.160	0.900	0.750	0.660	0.605	0.585	0.550			
0.70	1.350	1.300	0.753	0.660	0.600	0.566	0.542			
0.80	2.000	1.320	0.930	0.682	0.600	0.570	0.545			
0.90	3.000	2.000	1.200	0.800	0.625	0.590	0.555			
1.00	4.000	3.000	2.000	1.200	0.800	0.625	0.590			
*RPM _{idle} = 750, HP _{max} = 150, RPM _{max} = 4700, idle fuel flow = 10.0 lb/h										
POWER LIMIT CURVE DATA										
f_{HP}	0.384	0.540	0.728	0.860	0.940	0.985	1.00	0.967	0.900	0.790
f_{RPM}	0.120	0.200	0.300	0.400	0.500	0.600	0.700	0.800	0.900	1.00
f_{HP}	HP/HP _{max@rpm} , $f_{RPM} = RPM/RPM_{max}$									

Table 3-12. Fuel Consumption and Power Limit Curve Data
for Chrysler Upgraded (Metal) Gas Turbine

BRAKE SPECIFIC FUEL CONSUMPTION MAP DATA (lb/bhp-h)*										
f_{RPM}	f_{HP}									
	0.10	0.15	0.25	0.35	0.50	0.75	1.00			
0.10	2.000	1.700	1.300	1.100	0.900	0.750	0.670			
0.20	1.100	0.950	0.820	0.700	0.630	0.550	0.500			
0.30	0.900	0.790	0.650	0.580	0.535	0.492	0.480			
0.40	0.820	0.685	0.580	0.530	0.495	0.475	0.450			
0.50	0.800	0.640	0.540	0.500	0.485	0.462	0.440			
0.60	0.900	0.700	0.540	0.498	0.482	0.448	0.440			
0.70	1.100	0.900	0.550	0.500	0.482	0.447	0.440			
0.80	2.000	1.400	0.780	0.550	0.490	0.448	0.440			
0.90	2.000	2.000	1.300	0.900	0.550	0.472	0.440			
1.00	2.000	2.000	2.000	1.700	1.300	0.600	0.470			
*RPM _{idle} = 500, HP _{max} = 100, RPM _{max} = 4700, idle fuel flow = 4.17 lb/h										
POWER LIMIT CURVE DATA										
f_{HP}	0.305	0.636	0.883	0.974	1.00	1.00	1.00			
f_{RPM}	0.083	0.167	0.333	0.500	0.667	0.883	1.00			
f_{HP}	HP/HP _{max@rpm} , $f_{RPM} = RPM/RPM_{max}$									

Table 3-13. Fuel Consumption and Power Limit Curve Data for the Chrysler Upgraded (Ceramic) Gas Turbine

BRAKE SPECIFIC FUEL CONSUMPTION MAP DATA (lb/bhp-h)*

f_{RPM}	f_{HP}						
	0.10	0.15	0.25	0.35	0.50	0.75	1.00
0.10	1.750	1.487	1.137	0.962	0.787	0.656	0.586
0.20	0.962	0.831	0.717	0.612	0.551	0.481	0.438
0.30	0.787	0.691	0.569	0.507	0.468	0.430	0.420
0.40	0.717	0.599	0.507	0.464	0.433	0.416	0.394
0.50	0.700	0.560	0.472	0.438	0.424	0.404	0.385
0.60	0.787	0.612	0.472	0.436	0.422	0.392	0.385
0.70	0.962	0.787	0.481	0.438	0.422	0.391	0.385
0.80	1.750	1.225	0.682	0.481	0.529	0.392	0.385
0.90	1.750	1.750	1.137	0.787	0.481	0.413	0.385
1.00	1.750	1.750	1.750	1.487	1.137	0.525	0.411

*RPM_{idle} = 750, HP_{max} = 120, RPM_{max} = 4700, idle fuel flow = 3.0 lb/h

POWER LIMIT CURVE DATA

f_{HP}	0.305	0.636	0.883	0.974	1.01	1.01	1.00
f_{RPM}	0.083	0.167	0.333	0.500	0.667	0.833	1.00

$f_{HP} = HP/HP_{max@rpm}$, $f_{RPM} = RPM/RPM_{max}$

Table 3-14. Fuel Consumption and Power Limit Curve Data for the Kronogard KTT Gas Turbine Engine

BRAKE SPECIFIC FUEL CONSUMPTION MAP DATA (lb/bhp-h)*

f_{RPM}	f_{HP}						
	0.10	0.15	0.25	0.40	0.60	0.80	1.00
0.166	0.666	0.617	0.593	0.573	0.573	0.573	0.573
0.250	0.639	0.573	0.513	0.482	0.474	0.485	0.507
0.333	0.617	0.551	0.485	0.452	0.441	0.441	0.469
0.500	0.612	0.533	0.474	0.430	0.414	0.412	0.432
0.666	0.639	0.575	0.485	0.432	0.412	0.407	0.423
0.833	0.661	0.639	0.524	0.456	0.423	0.414	0.425
1.00	0.880	0.771	0.617	0.507	0.445	0.430	0.436

*RPM_{idle} = 750, HP_{max} = 150, RPM_{max} = 6000, idle fuel flow = 5.0 lb/h

POWER LIMIT CURVE DATA

f_{HP}	0.299	0.611	0.860	0.955	1.00	1.00	0.980
f_{RPM}	0.083	0.167	0.333	0.500	0.666	0.833	1.00

$f_{HP} = HP/HP_{max@rpm}$, $f_{RPM} = RPM/RPM_{max}$

Table 3-15. Fuel Consumption and Power Limit Curve Data for the
Garrett Single-Shaft (Metal) Gas Turbine

BRAKE SPECIFIC FUEL CONSUMPTION MAP DATA (lb/bhp-h)*							
f_{RPM}	f_{HP}						
	0.05	0.15	0.30	0.50	0.70	0.85	1.00
0.40	1.200	0.900	0.700	0.550	0.460	0.420	0.410
0.50	1.200	0.900	0.620	0.500	0.450	0.420	0.410
0.60	1.000	0.670	0.550	0.490	0.450	0.420	0.410
0.70	1.200	0.800	0.580	0.480	0.430	0.420	0.410
0.80	1.200	0.800	0.550	0.460	0.420	0.415	0.410
0.90	1.200	0.900	0.580	0.460	0.420	0.420	0.390
1.00	1.200	0.820	0.630	0.480	0.460	0.450	0.410
* $RPM_{idle} = 2000$, $HP_{max} = 120$, $RPM_{max} = 4700$, idle fuel flow = 5.0 lb/h							
POWER LIMIT CURVE DATA							
f_{HP}	0.04	0.11	0.19	0.31	0.51	0.76	1.00
f_{RPM}	0.50	0.57	0.60	0.70	0.80	0.90	1.00
$f_{HP} = HP/HP_{max@rpm}$, $f_{RPM} = RPM/RPM_{max}$							

Table 3-16. Fuel Consumption and Power Limit Curve Data for the
Garrett Single-Shaft (Ceramic) Gas Turbine

BRAKE SPECIFIC FUEL CONSUMPTION MAP DATA (lb/bhp-h)*							
f_{RPM}	f_{HP}						
	0.05	0.15	0.30	0.50	0.70	0.85	1.00
0.4	1.05	0.787	0.612	0.481	0.402	0.367	0.359
0.5	1.05	0.787	0.542	0.438	0.394	0.367	0.359
0.6	0.875	0.586	0.481	0.429	0.394	0.367	0.359
0.7	1.05	0.700	0.507	0.420	0.376	0.367	0.359
0.8	1.05	0.700	0.481	0.402	0.367	0.363	0.359
0.9	1.05	0.787	0.507	0.402	0.367	0.367	0.341
1.0	1.05	0.717	0.551	0.420	0.402	0.394	0.359
* $RPM_{idle} = 2000$, $HP_{max} = 120$, $RPM_{max} = 4700$, idle fuel flow = 3.5 lb/h							
POWER LIMIT CURVE DATA							
f_{HP}	0.04	0.11	0.19	0.31	0.51	0.76	1.00
f_{RPM}	0.50	0.57	0.60	0.70	0.80	0.90	1.00
$f_{HP} = HP/HP_{max@rpm}$, $f_{RPM} = RPM/RPM_{max}$							

Table 3-17. Idle Fuel Flow and Effective Rotational Inertia for Various Automotive Gas Turbine Engines

Engine	HP	Idle Fuel Flow, lb/h	Effective ⁽¹⁾ Rotational Inertia, lb-in. ²
Chrysler Baseline	150	10.0	2500
Chrysler Upgraded (metal)	97	4.2	1390
	49	2.5	665
Chrysler Upgraded (ceramic)	97	2.9	485
	49	1.8	332
Kronogard KTT	150	5.0	1900
Garrett single-shaft (metal)	90	4.0	1290
	46	2.4	635
Garrett single-shaft (ceramic)	90	2.8	355
	46	1.9	255

$$^{(1)} I_{\text{eff}} = (\text{Reduction Gear Ratio})^2 I_{\text{engine}}$$

on the Chrysler Baseline engine indicate that the low emissions goals (0.4 gm/mi HC, 3.4 gm/mi CO, 0.4 gm/mi NO_x) can be met using an appropriate fuel control system. There are no emissions data available for combustors intended for use with ceramic components and the associated turbine inlet temperatures of up to 2500°F. Meeting the 0.4 gm/mi NO_x standard with combustors having primary reaction zone temperatures of 2400-2600°K or higher could prove to be difficult (see Reference 3-18).

Since nearly all the meaningful emissions data in the literature are given in connection with vehicle tests, the quantitative discussion of gas turbine emissions will be presented in Section 3.4.4.5 in terms of vehicle emissions on the Federal Driving Cycle.

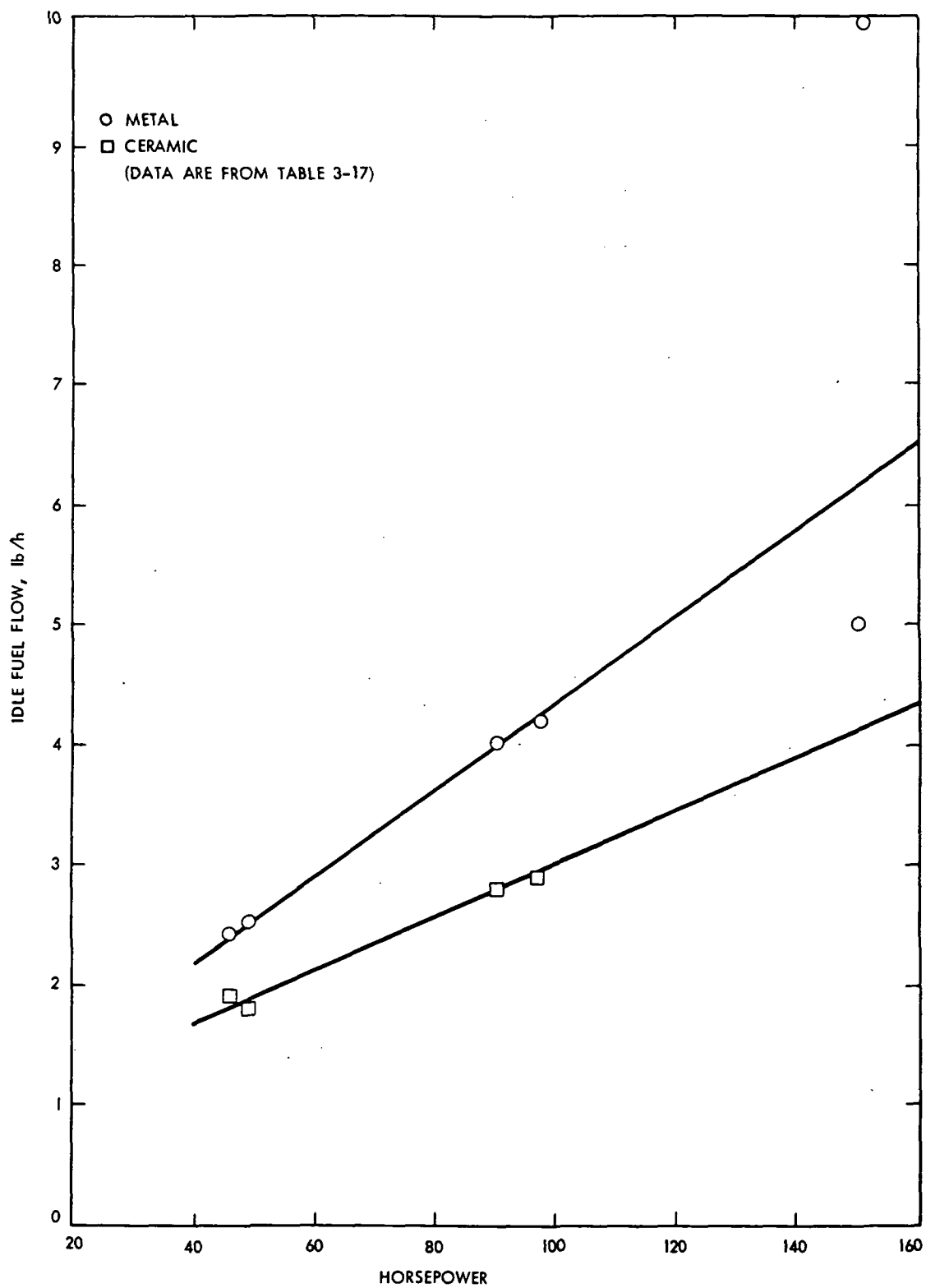


Figure 3-56. Idle Fuel Flow for Automotive Gas Turbine Engines

3.4.4 Characteristics of Gas Turbine-Powered Passenger Cars

3.4.4.1 Vehicle Programs. Gas turbine engines have been installed and tested in passenger cars for about 30 years (Reference 3-2), starting with Rover in 1949 and Chrysler in 1954. Recent gas turbine passenger car programs (1970-78) are summarized in Table 3-18. Considerable data/information are available concerning the programs listed in the table and that data base is used for much of the discussion given in the following subsections.

3.4.4.2 Packaging. Gas turbine engines have been successfully installed in various size passenger cars. The Chrysler (Figure 3-57) and GM projects have involved intermediate and full-sized cars with the replacement of conventional L6 or V8 engines. The Volkswagen/Williams programs have involved the replacement of L4 and rotary engines and thus have been concerned with smaller engine compartments. The installation by Volkswagen of the GT-150 engine in the front-wheel drive Ro-80 (Figure 3-58) is particularly significant. Further evidence of the packagability of a gas turbine (KTT-MKI) in a relatively small car is shown in Figure 3-59. Hence there seems little question that gas turbine engines of 100-150 hp can be installed effectively in the downsized intermediate or full-sized cars of the 1980s. Whether or not gas turbines of the 40-60 hp size can be packaged into the same space as conventional engines of comparable horsepower is not yet clear. In addition, questions regarding the effect of passenger car packaging constraints on engine efficiency (bsfc) must be resolved. Nevertheless, available information indicates that packaging constraints should not prevent the use of gas turbine engines in passenger cars. From the standpoint of packaging, incorporation of the transmission into the gas turbine may be a distinct advantage.

3.4.4.3 Performance and Driveability. Considerable experience has been gained over the years on the performance and driveability of gas turbine-powered cars. All the on-the-road experience was obtained with two-shaft (free turbine) engines as no single-shaft engines have been built to date which are suitable for vehicle testing. Except for a response lag at the initiation of an acceleration and inadequate engine braking for some engine designs, the driving characteristics of gas turbine-powered cars have been found to be quite acceptable. The quietness and smoothness of the engine are particularly noteworthy. Questions concerning acceleration response and engine braking are discussed in the following paragraphs.

The acceleration response lag of the gas turbine is caused by the need for high gasifier (compressor) RPM to produce high power. The effective rotational inertia of the gasifier system, including any accessories which may be driven by it, is sufficiently large that the time to accelerate it from idle speed to rated speed (with no external load) is 1.0-1.5 seconds even for a well-designed engine. For example, the Chrysler Baseline engine has a gasifier response time of about 1.3 second and the Upgraded engine has a response time of 1.1 seconds.

Table 3-18. Gas Turbine Passenger Car Programs (1970-78)

Year	Company	Engine	HP	Vehicle	Type of Tests	Status	Reference
1974	Chrysler	Baseline	150	Plymouth Satellite	Road and dynamometer	Completed	3-21
1977	Chrysler	Upgraded	105	Plymouth Aspen	Engine and dynamometer	Continuing	3-5, 3-32, 3-34, 3-35, 3-36
1975	GM	GT-225	225	Chevy Impala	Road and dynamometer	Completed	3-19
1971	Volkswagen/ Williams	GT-70	75	VW Microbus, VW 1600	Road and dynamometer	Completed	3-2, 3-4, 3-31
1977	Volkswagen/ Williams	GT-150	135	NSU-Ro80	Road and dynamometer	Continuing	3-4
1972	Williams	WR 26	80	AMC Hornet	Road and dynamometer	Completed	3-2
1978	United Turbine	KTT-MKI	100	Volvo	Engine dynamometer	Continuing	3-3, 3-11, 3-12, 3-13, 3-14, 3-28
1978	GM	AGT-5	125	Olds 88	Road and dynamometer	Continuing	3-59

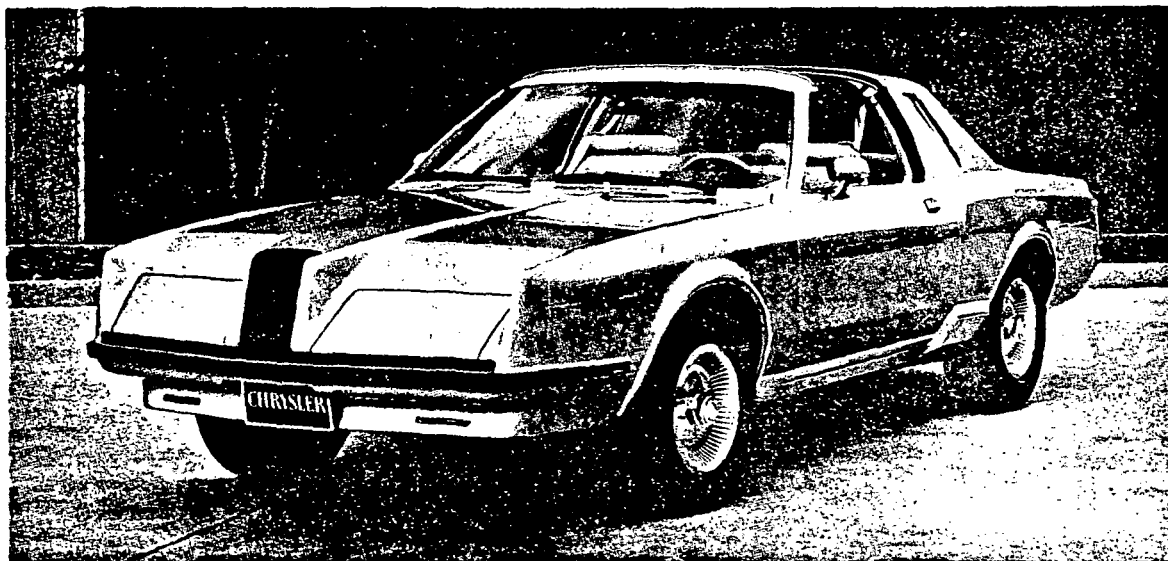
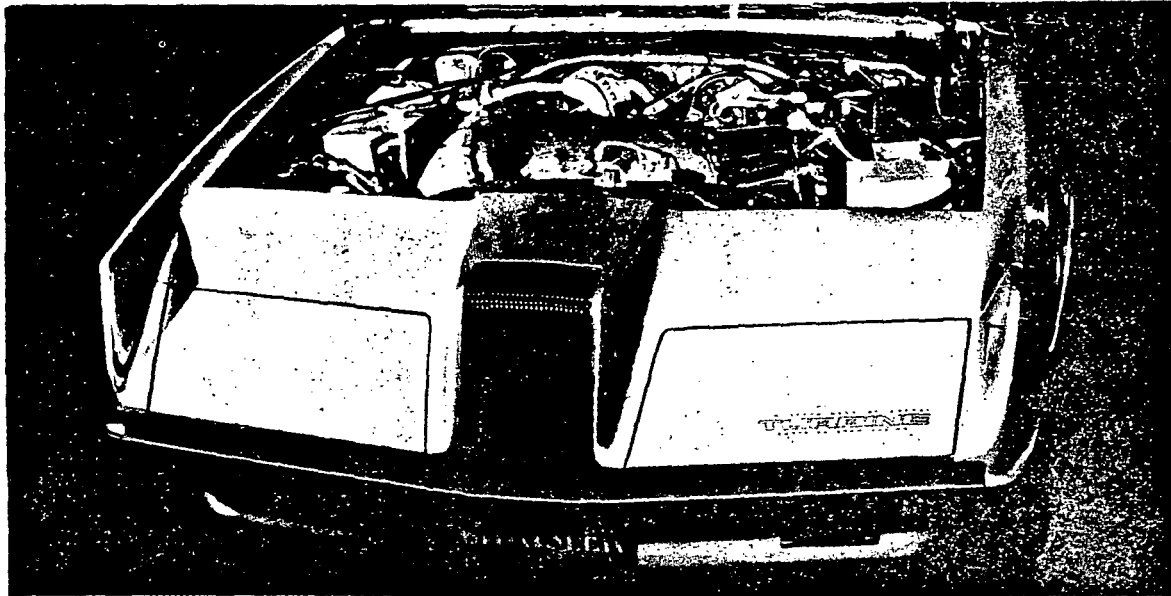


Figure 3-57. Chrysler Upgraded Gas Turbine in a Turbine Concept Car
Based on the Chrysler Le Baron (Reference 3-34)

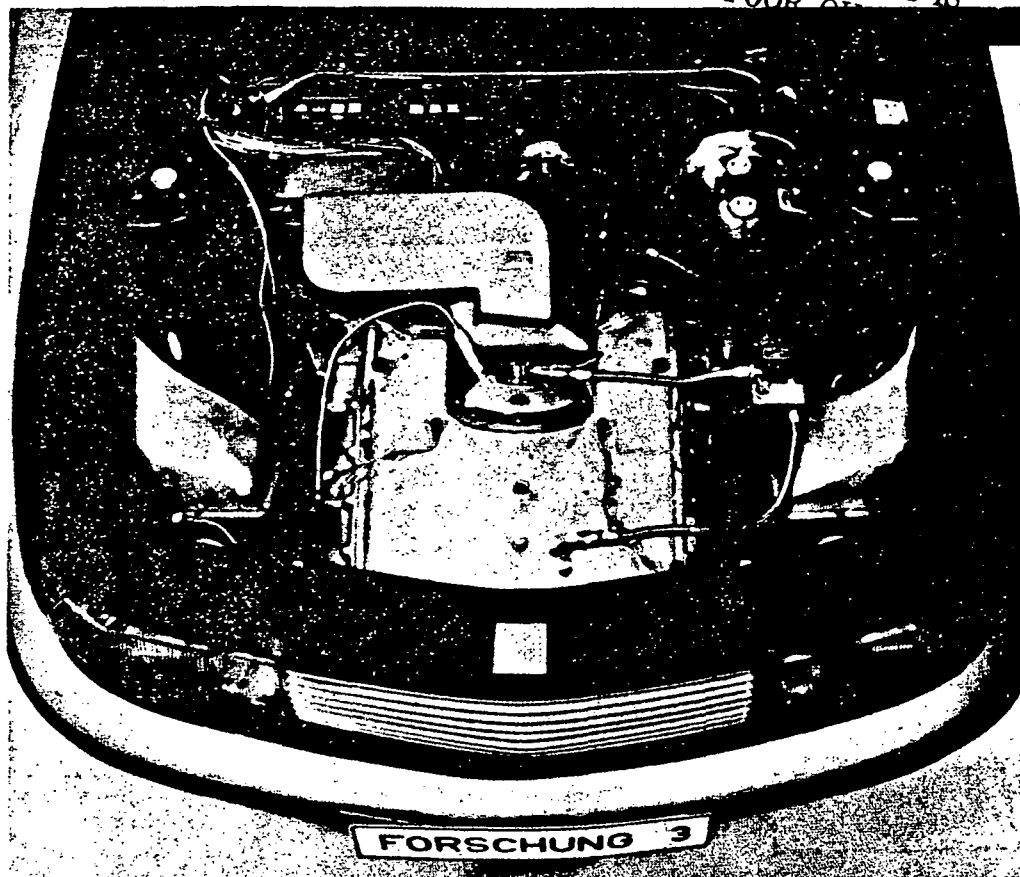


Figure 3-58. VW GT-150 in the Ro-80 (Reference 3-4)

Single-shaft ceramic engines have projected response times of 0.8-1.0 seconds (Reference 3-8). Performance data for recent gas turbine-powered passenger vehicles are given in Table 3-19.

Some of the earlier gas turbine-powered cars had less than adequate engine braking, but recent designs have exhibited satisfactory performance in that regard. Engine braking can be implemented by either reversing the torque on the power turbine using variable-inlet nozzles or by extracting more power in the compressor than is supplied from the gasifier turbine. The first approach is applicable only to multishaft engines while the second is applicable to both single and multishaft engines. In the latter cases, the gasifier and power shafts must be clutched together during braking. Engine braking for gas turbine engines can be tailored to match almost any vehicle braking requirements by appropriate control of engine geometry and fuel flow.

3.4.4.4 Fuel Economy. The fuel economy of gas turbine-powered passenger cars has been the subject of considerable interest and activity in recent years. All design studies (References 3-7, 3-8, 3-9) of gas

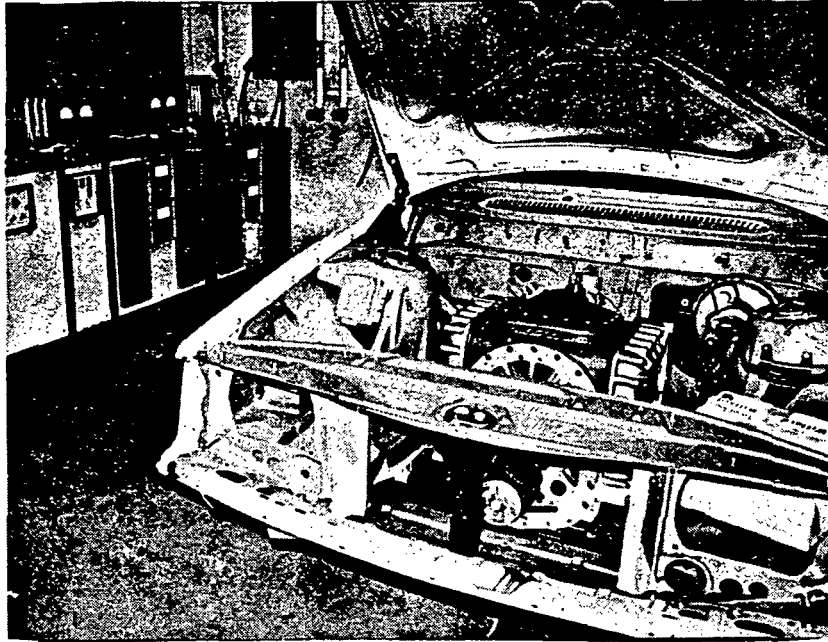


Figure 3-59. Mark I (100 HP) KTT Engine in a Compact Car (Reference 3-28)

turbine engines have focused on fuel economy as a key characteristic of the engine and dynamometer and road tests of prototype engines in vehicles have emphasized reliable measurements of fuel economy at an early date in the test program. Hence there is available in the literature considerable information on the fuel economy of gas turbine-powered cars. In addition, as part of the present study at JPL, computer simulations of the fuel economy of various engine/vehicle combinations were made as an aid in comparing the fuel economy potential of the various engines on a consistent basis. Some of the JPL computer simulation results for gas turbine vehicles are given in Table 3-20.

Constant speed fuel economy using gas turbine engines is shown in Figures 3-60 and 3-61 for speeds between 15 and 75 mph. The fuel economy is expressed as (mpg times kilopounds) so that data/calculations for cars of different inertia weights can be compared on the same plot. The results shown in Figure 3-59 for the Chrysler Baseline, VW GT-70, and GM GT-225 engines are based on actual experimental data taken in road and/or chassis dynamometer tests. The remaining curves on Figures 3-60 and 3-61 are based on engineering calculations and reflect different assumptions relative to component efficiencies and engine operating conditions. These curves show the wide range of the projections of constant-speed fuel economy for gas turbine-powered passenger cars. Since much passenger car operation occurs under transient conditions, passenger car fuel economy strongly depends on the transient characteristics of the engine. Volkswagen has made a rather extensive series (References 3-29, 3-32, 3-38) of studies about the transient operation of gas turbine engines.

Table 3-19. Performance Data for Gas Turbine-Powered Passenger Cars

Developer	Engine ⁽¹⁾	hp	Vehicle	Inertia Weight, lb	hp/lb	Response Time, sec	Acceleration 0-60 mph, sec	Distance 10 sec, ft
GM	GT-225	225	Chevy Impala	5600	0.04	1.7	11.9	--
Williams	WR 26	80	AMC Hornet	3150	0.025	1.7	18	--
VW/Williams	GT-70	75	VW 1600	2900	0.026	1.4	14.4	--
			VW Microbus	3500	0.021		16.5	--
	GT-150	136	Ro80	3700	0.036	---	14	426
Chrysler	Baseline	150	Plymouth Satellite	4500	0.033	1.3	12	448
	Upgraded	120 (Augm.)	Plymouth Aspen	3600	0.033	1.1	13.5	--

(1) All vehicles use three-speed automatic transmission with a torque converter.

Table 3-20. Summary of JPL Computer Results for Fuel Economy of Gas Turbine-Powered Passenger Cars

Engine	Iw	hp	TRM	CD A, ft ²	Wt, lb/h	Fuel Economy, mpg ⁽¹⁾		
						City	Hwy	Composite
Chrysler Baseline	3500	150	M4	10	10	10.2	22.0	13.0
VW GT-70	3500	70	M4	15	10	10.3	17.6	12.6
Kronogard KTT	3500	150	M4	10	5	17.4	35.0	22.4
Chrysler Upgraded								
Metal	3500	120	M4	11	5	15.4	30.9	19.8
Metal	3500	120	CVT	11	5	14.8	28.0	18.7
Metal	3500	100	M4	11	4.7	18.8	33.0	23.3
Ceramic	3500	120	M4	11	3.5	18.9	35.6	24.0
Ceramic	3500	100	M4	11	2.9	21.3	37.6	26.5
Metal	2050	50	M4	8.5	2.5	32.9	38.8	38.5
Ceramic	2050	50	M4	8.5	2.0	37.1	55.7	43.7
Garrett Single-Shaft								
Metal	3500	120	CVT	11	5	17.6	36.5	22.9
Metal	3500	120	CVT	11	3	20.8	37.2	25.9
Ceramic (2)	3500	120	CVT	11	5	19.7	41.6	25.8
Ceramic (2)	3500	120	CVT	11	3	22.7	42.1	28.6
Metal	3500	100	CVT	11	4.2	19.2	39.0	24.9
Ceramic (2)	3500	100	CVT	11	2.9	23.4	44.8	29.8
Metal	2050	50	CVT	8.5	2.5	33.0	50.3	39.0
Ceramic (2)	2050	50	CVT	8.5	2.0	37.6	57.8	44.6

(1) Diesel fuel - density - 7.2 lb/gal

(2) Ceramic version of the metal engine - not advanced Garrett single-shaft ceramic engine in Table 3-21.

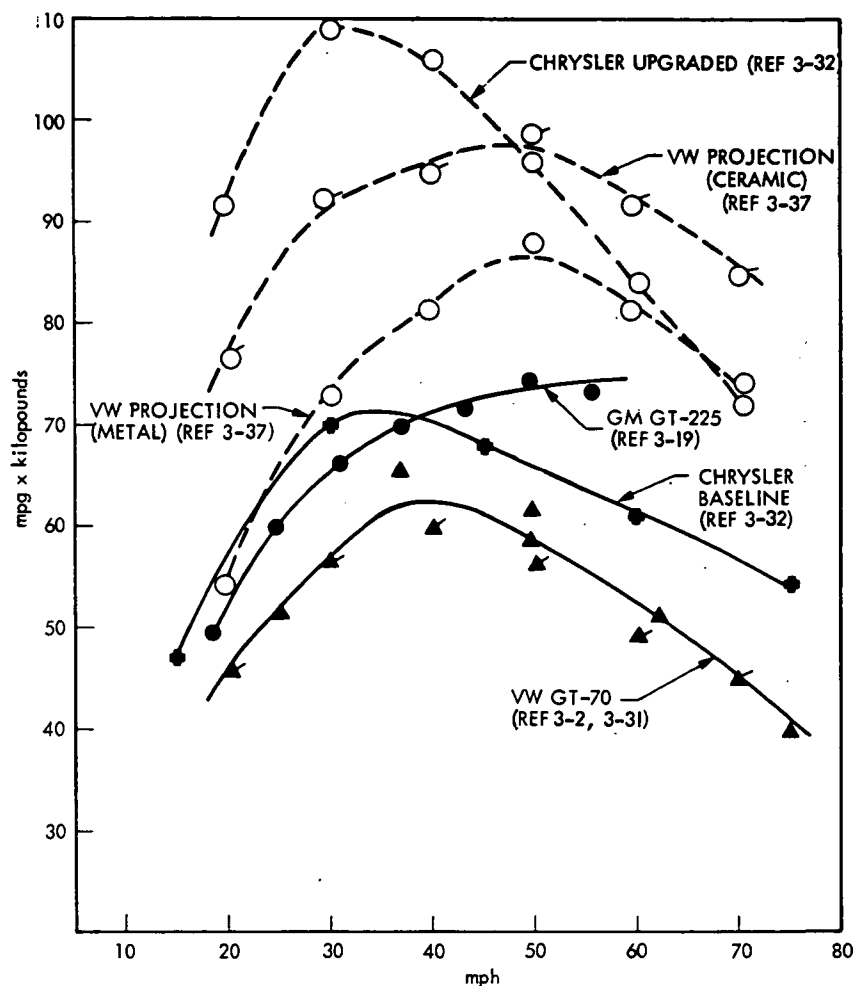


Figure 3-60. Constant-Speed Fuel Economy for Gas Turbine-Powered Passenger Cars

Fuel economy data for urban and highway driving cycles are available for the VW GT-70 and GT-150 engines (References 3-4, 3-39) and the Chrysler Baseline engine (Reference 3-35). Data for the VW GT-70 in a typical passenger car are shown in Table 3-21 for urban, suburban, and highway driving. For 100 miles of driving, the vehicle had a composite mileage of 11.5 mpg. Later reports indicate that the VW GT-150 achieved a composite fuel economy of 14 mpg in the Ro-80 vehicle (inertia weight of 3700 lb). The Chrysler Baseline engine in a Plymouth Satellite (inertia weight of 4500 lb) was tested extensively on a chassis dynamometer by EPA (Reference 3-35). The results of these tests indicated a city fuel economy of 6.8 mpg and a highway fuel economy of 12.9 mpg. It is expected that when the development modifications are completed on the Chrysler Upgraded engine, significant progress will be made toward demonstrating the fuel economy potential of the gas turbine engine in passenger cars.

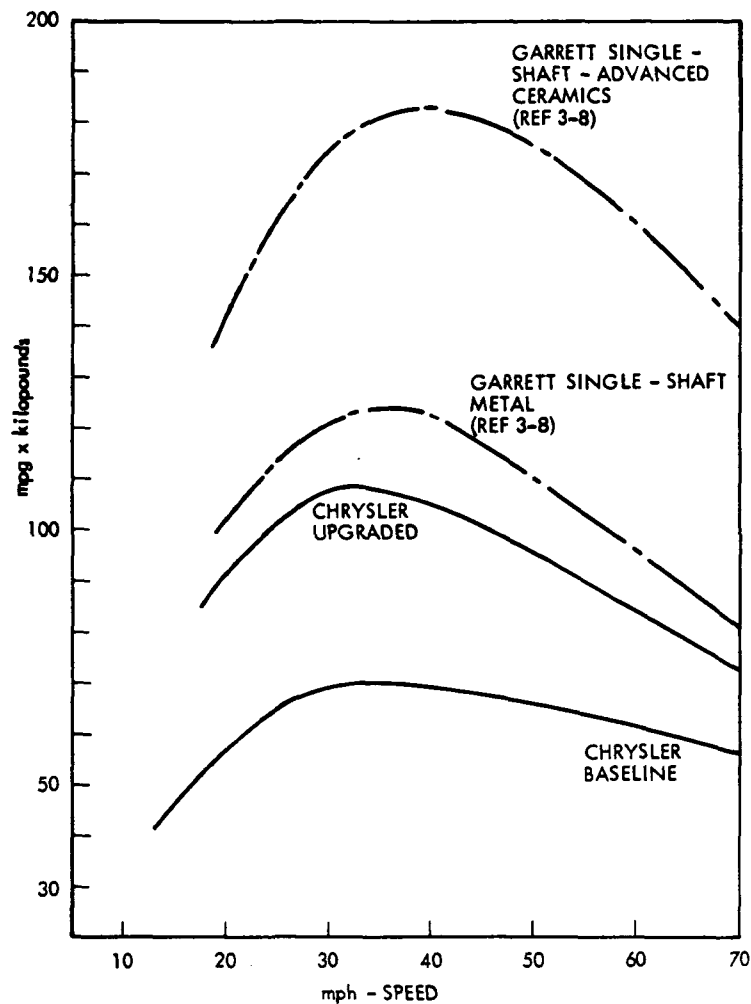


Figure 3-61. Projected Constant-Speed Fuel Economy for Gas Turbine-Powered Passenger Cars

3.4.4.5 Emissions. Considerable information is available concerning emissions from gas turbine-powered cars over the Federal Driving Cycle. Comprehensive discussions of the design and development of operating strategies and control systems for combustors to meet strict vehicle emissions standards are given in References 3-4, 3-19, 3-21. That significant progress has been made in recent years is shown by the data given in Table 3-22. The data given in the table indicate that the 0.4 g/mi HC and 3.4 g/mi CO standards can be met with several combustor designs including the simple diffusion flame, single-can combustor, but a NO_x standard of less than 2.0 g/mi NO_x requires the use of a more complex design which utilizes a premixed, vaporized fuel with possibly more than one stage of burning. As indicated in Figure 3-62, it is very likely that combustors can be developed to meet the standards of 0.4 g/mi HC, 3.4 g/mi CO, and 0.4 g/mi NO_x . Such a combustor has not yet been demonstrated in a vehicle having satisfactory driveability, but there is little doubt that it can be done. The major question is concerned with the need or desirability of using variable geometry in such a combustor.

Table 3-21. Typical Passenger Car Fuel Economy Using the VW GT-70
(Reference 3-39)

Mode	% Time	Mean Speed, mph	Distance, mi	Diesel ⁽¹⁾ Oil, gal	Mileage, mpg
FDC	50	19.84	33	4.49	7.4
Suburban	33	30	33	1.91	17.3
Highway	17	60	34	2.31	14.7
Total			100	8.71	11.5

(1) Fuel density = 6.8 lb/gal.

All of the emissions data cited in Table 3-22 are for two-shaft gas turbines using turbine inlet temperatures in the range of 1800-1900°F. Little work has been reported on emissions from single-shaft engines or from combustors intended for use with ceramic components (TIT up to 2500°F). There is no reason to believe that exhaust emissions from a metal single-shaft engine should be much different than those from a comparable two-shaft engine. Hence, available emissions data should be applicable to single-shaft engines as well. The effect of increased turbine inlet temperature on emissions, especially NO_x, is more difficult to assess. It is to be expected that HC emissions will be even lower at high TIT because the potential surface quench zones will be hotter. Reduction of CO emissions will be favored by the higher reaction zone temperatures, but the lower overall air/fuel ratios (less lean) of the combustor will be unfavorable. The primary question regarding emissions for vehicles using ceramic gas turbines is, however, concerned with NO_x emissions. Turbine inlet temperatures of 2500°F will result in reaction zone temperatures at least several hundred degrees higher. Increasing combustor temperatures are certainly unfavorable from a NO_x emission standpoint. NO_x formation could even be a problem in premixed, gaseous (vaporized) fuel combustors such as catalytic combustors at the high temperatures projected for ceramic gas turbine operation. Little information is currently available in this area.

3.5 CRITICAL PROBLEMS

3.5.1 Introduction

Many aspects of the design, development, and status of automotive gas turbine engines and gas turbine-powered passenger cars have

Table 3-22. Summary of Turbine-Powered Vehicle Exhaust Emissions for Various Engines and Combustor Designs

Developer	Engine	Combustor	Vehicle/ Inertia Weight	Test Cycle	Fuel	Emissions, g/mi		
						HC	CO	NO _x
Chrysler	Baseline	Diffusion flame, single can	Plymouth/4500	FDC(1)	Diesel	0.68	3.6	2.8
	Baseline	Premixed, torch ignition	Plymouth/4000	FDC	Gasoline	2.5	8.6	0.44
VW	GT-70	Toroidal	Microbus	FDC	JP-4	0.5	4.9	2.6
	GT-70	Toroidal	VW 1600	FDC	Gasoline	0.35	4.5	2.2
VW	GT-150	Diffusion flame, single can	Ro-80/3700	FDC	--	<0.4	<3.4	≥2.0
	GT-150	Premixed, single stage	Ro-80/3700	FDC	--	<0.4	<3.4	>1 but <2
GM	GT-225	Diffusion flame, single can	Impala/5000	FDC	Kerosene	0.20	2.8	6.0
	GT-225	Prechamber, premixed, prevaporized	Impala/5000	FDC	Diesel	0.18	2.0	0.38
Williams	WR26	Toroidal	Hornet/3000	FDC	Gasoline	0.27	6.8	2.9

(1) Cold start, 1975 test procedure in most cases.

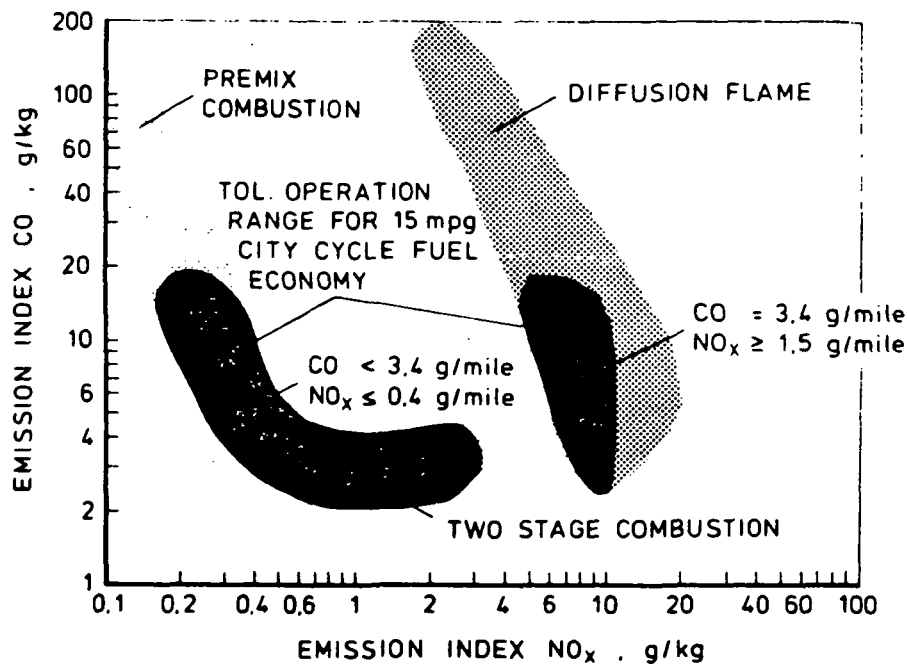


Figure 3-62. CO-NO_x Ranges for Different Combustion Systems

been discussed above in detail. It is clear from the material presented that much progress has been made in the last 5-10 years on the development of a gas turbine engine suitable for use in a passenger car, but that much more progress is required before the gas turbine approaches the potential projected for it in the passenger car application. In what follows, the critical problems remaining are identified. As might be expected, nearly all the critical problems cited are currently the subject of significant activity in the DOE Gas Turbine Program (see Tables 3-4 and 3-5).

Two of the major critical areas for the automotive gas turbine are fuel economy and production cost. Significant progress must be made in improving fuel economy and reducing the projected production cost of the gas turbine before its recognized advantages in the areas of low emissions, multifuel capability, smoothness, and quietness are likely to be realized. Hence, in one way or another, all of the critical problems discussed are related to fuel economy or production cost. The discussion of critical problems must also recognize the fact that the gas turbine is part of a vehicle system and, because of this, there must inevitably be trade-offs between the design of the gas turbine itself and the vehicle. For this reason, the discussion of the critical problems is divided into separate considerations of engine-related and vehicle-related problems. Those problems of gas turbine development which are most closely tied to engine-vehicle trade-offs are included in the vehicle-related problems subsection.

3.5.2 Engine-Related Problems

At or near their design point (usually near rated power), gas turbine engines can have high efficiency (low bsfc). Most of the engine-related critical problems are closely associated with attaining a low bsfc at the design point and then maintaining efficient operation over a sufficiently wide range of load and RPM. This requirement for efficient operation over a wide range of conditions has made even the selection of the optimum engine and shafting configuration difficult. As discussed in Section 3.2, there are a multiplicity of possible configurations. Hence the initial critical problem is the selection of one or at most several (two or three) component/shafting configurations which are best suited for development as passenger car gas turbine engines. This selection process must be done systematically and should recognize the consequences of size-effects and the probable need for ceramic components to meet fuel economy goals. Regardless of the engine configurations selected, there will be a need for variable geometry, increased turbine inlet temperature, and improved cycle heat exchangers (higher heat recovery and lower leakage). Hence the critical engine-related problems are as follows:

- (1) Selection of engine configurations best suited for development for automotive applications.
- (2) Development of variable-geometry components (compressor inlet guide-vanes, turbine nozzles, and diffusers).
- (3) Development of ceramic materials and components (stationary-fixed, stationary-variable, rotating stressed).
- (4) Improvement of regenerators (or recuperators).
- (5) Minimization of size-effects on component efficiencies by appropriate design and improved manufacturing processes.

All of these problems must be dealt with successfully if the gas turbine is to be attractive from a fuel economy point-of-view. The solutions must also be such that the production cost of the gas turbine is not significantly greater than that of the conventional engines of the same time period.

3.5.3 Vehicle-Related Problems

The critical vehicle-related problems are concerned with the trade-offs between engine/component design and packaging the engine in a vehicle and matching the engine torque/power characteristics with vehicle requirements for road load and accessories. In the engine design and packaging area, the critical problems are identifying and understanding the trade-offs between engine efficiency, weight and volume, and how the

overall vehicle system consequences (e.g., vehicle weight and fuel economy) of the trade-offs can best be handled. In the engine output/vehicle load matching area, the critical problems are understanding the trade-offs involved with providing the transmission function and how the engine/transmission can best be integrated together from a vehicle systems point-of-view.

There has been a tendency to consider the gas turbine as an add-on to an existing vehicle (i.e., to fit the engine into the engine compartment of a production vehicle). In addition, most gas turbine engines have been mated to a three-speed automatic transmission even though it is well recognized that the two-shaft engine has an inherent high torque at low RPM and does not require the torque converter. Further, the unavailability of a continuously variable transmission (CVT) has been a primary reason that single-shaft gas turbines have not been developed and tested in passenger cars. The critical aspects of these problems are that the gas turbine has often been used in a less-than-optimum powertrain and forced to fit into a prescribed engine compartment without a clear definition or determination of the associated engine performance penalties.

The critical vehicle-related problems discussed in the previous paragraphs can be stated as follows:

- (1) Understanding the effect of engine efficiency and packaging trade-offs on vehicle weight, style, performance, and fuel economy.
- (2) Understanding of the trade-offs between engine configuration and shafting, transmission requirements and efficiency, and vehicle performance.
- (3) Optimum integration of the engine and transmission into a powertrain unit.

All of these problems must be understood and addressed before the attractiveness of the gas turbine for the downsized vehicles of the 1980s can be properly assessed and the most advantageous engine configuration selected.

3.6 RECOMMENDATIONS FOR FUTURE R&D

As noted in the previous section, the critical problem areas for the gas turbine are fuel economy and production cost. In addition, it was noted that it is appropriate to identify the critical problems as primarily engine-related or intrinsically connected with the design of the total vehicle/engine system. This same approach will be followed in this section, in which recommendations for future R&D on gas turbine engines for passenger cars will be discussed. The discussion will emphasize the identification of R&D topics or projects and will not treat in detail the objectives or approach that should be followed in each area. As would be expected, R&D efforts are presently under way

in many of the areas identified. In some cases more effort seems to be appropriate, especially in the vehicle-related critical problems areas.

3.6.1 Engine-Related R&D

It is convenient to discuss the engine-related R&D recommendations in terms of engine systems and component development.

3.6.1.1 Engine Systems. In the area of engine systems, it is recommended that the following R&D topics be covered:

- (1) Selection of at most two or three engine/shafting configurations best suited to passenger car applications.
- (2) Modular engine design and testing to evaluate component alternatives.
- (3) Design and testing of a state-of-the-art single-shaft engine at an early date.

Most of the past R&D effort on passenger car gas turbines has been concerned with the design and testing of the regenerative two-shaft engine with minimal study of alternative shafting arrangements. In addition, the engines built have been totally integrated packages for mounting in production cars. Alteration of those engines to evaluate different component designs has proven to be very time consuming and costly and has resulted in a scarcity of engine or vehicle data on which to base engine configuration or component selection. Available fuel economy data from gas turbine-powered passenger cars seems to indicate that engines utilizing the simplest two-shaft designs will have difficulty competing with the improving conventional engines, and a new and broader look at gas turbine design alternatives is needed. Such a new look has been initiated by DOE (e.g., three industry study team contracts currently in progress) during the last year. The engine system R&D items listed are intended to expedite the needed relook from both an analytical and hardware point-of-view.

3.6.1.2 Component Development and Evaluation. In the areas of component development and evaluation, it is recommended that the following R&D topics be covered:

- (1) Development of variable-geometry components (compressor inlet guide-vanes, compressor diffuser, turbine inlet nozzles).
- (2) Design for optimum use of ceramic materials.
- (3) Development of ceramic components (stationary-fixed, stationary-variable, rotating stressed).

- (4) Evaluation of air cooling as an alternative to ceramics (e.g., laminated air-cooled rotors).
- (5) Improvement of regenerative heat exchangers (higher heat recovery, lower leakage, comparison of advanced regenerators and recuperators).
- (6) Determination of size-effects on component and engine efficiencies for small hp engines and high turbine inlet temperatures (effect of design philosophy and manufacturing limitations).
- (7) Development of a low NO_x emissions combustor for use in a ceramic engine (TIT = 2500°F).

Component development and evaluation are certainly important parts of the current DOE gas turbine program. Present efforts seem to emphasize the improvement of rotating ceramic regenerator disks and the development of ceramic materials and components as the primary means of improving fuel economy and reducing production costs. Available information indicates that, even when using ceramics (TIT = 2500°F), variable-geometry capability is critical to achieving the part-load and idle fuel consumption that is required for passenger car engines. In addition, the results of the JPL study of size-effects in high temperature 50-100 hp engines (see Section 3.8) indicate that such effects are likely to be significant and could be very important in determining the viability of gas turbines in downsized passenger cars. Hence it is recommended that increased efforts be directed toward the development of variable-geometry components, especially ceramic ones, and that studies to assess the importance of size-effects and ways of minimizing their impact be emphasized. The need to minimize the size-effects in high temperature engines is of critical importance, especially as it influences manufacturing processes and production costs.

3.6.2 Vehicle-Related R&D

There are some aspects of gas turbine engine design and development which are intrinsically connected with the design and operation of the vehicle (passenger car) in which the engine is to be used. Recommended R&D activities in those areas are listed below.

- (1) Studies to determine the trade-offs between engine packaging, efficiency, and cost and vehicle-weight, performance, and fuel economy.
- (2) Studies to optimize the engine/transmission unit configuration from the vehicle integration, performance, fuel economy, and cost points-of-view.
- (3) Determination of continuously variable transmission (CVT) requirements for use with a single-shaft engine.

- (4) Study of the use of transmission/energy storage combinations to load level both the two-shaft and single-shaft engines.
- (5) Study of the trade-offs between engine and vehicle response, idle fuel flow, and fuel economy and their relation to engine design philosophy.
- (6) Study of relationships between engine braking, deceleration fuel flow, engine response, and vehicle fuel economy.
- (7) Development of engine/transmission/vehicle simulation models and computer programs which realistically account for the effect of transients on vehicle performance and fuel economy.

At the present time, the DOE Gas Turbine Program seems to have minimum efforts in most of the vehicle-related R&D areas listed above. Significant additional efforts in these areas are needed in support of the direct engine-related R&D activities which presently make up the DOE Gas Turbine Program. Vehicle-related studies, such as those identified, would contribute importantly to the optimization of gas turbine power-trains for use in passenger cars and the improvement of vehicle fuel economy for driving cycles and operating modes typical of such applications.

3.7 SUMMARY AND CONCLUSIONS

Various aspects of the development of gas turbine engines and their use in passenger cars have been discussed in considerable detail in this section. The essence of those discussions is summarized in this subsection in terms of a number of conclusions subdivided as follows: (1) present status, (2) projected development, (3) critical problems, and (4) recommended R&D.

3.7.1 Present Status

- (1) The DOE Gas Turbine Program is a coordinated effort to demonstrate that gas turbines can be developed which would be attractive for use by the automotive industry in passenger cars. Participation in the program has been expanded in recent years to include groups/companies which have been active in the past in heavy duty truck and aircraft gas turbine development. The focal point of the DOE program has been the Chrysler Upgraded Engine (two-shaft, 105 hp) which is currently undergoing development modifications and dynamometer testing. More recently program activity has begun to shift to an emphasis on ceramic materials and component fabrication and definition of engine configurations for future development.

- (2) There are passenger car gas turbine programs outside the United States which have made significant progress in the last 5 years. These include the programs at Volkswagen (West Germany) and United Turbine (Sweden). VW (in conjunction with Williams Research, USA) has developed a state-of-the-art two-shaft engine (GT-150) and tested it in a Ro-80 car. United Turbine has designed and is currently testing a three-shaft (KTT) engine on a dynamometer.
- (3) The fuel economy of the Ro-80 car ($I_w \approx 3500$ lb) powered by the VW GT-150 (136 hp) was found to be 14 mpg for the EPA composite driving cycle. This fuel economy was a significant improvement over that found using earlier gas turbine designs (e.g., Chrysler Base-line and the VW GT-70), but it is still significantly less than that for 1978 production passenger cars using conventional V-8 engines (16-18 mpg composite cycle, California emission standards - 0.4 g/mi HC, 9.0 g/mi CO, 1.5 g/mi NO_x).
- (4) The Ro-80 car with the VW GT-150 gas turbine engine (diffusion flame combustor) met emission standards of 0.4 g/mi HC, 3.4 g/mi CO, and 2.0 g/mi NO_x. Engines (GM GT-225) utilizing experimental premixed, vaporized fuel combustors have met the research goal standards of 0.4 g/mi HC, 3.4 g/mi CO, and 0.4 g/mi NO_x, but the vehicles exhibited driveability problems.

3.7.2 Projected Development

- (1) Most current projections of future automotive gas turbine engines envision the use of single-shaft configurations and ceramic components. No single-shaft engines suitable for use in a passenger car have been built to date, primarily because of the unavailability of a wide speed-ratio range, high efficiency continuously variable transmission (CVT). Development of the technology to fabricate ceramic components, especially the turbine rotor, is in a relatively early stage.
- (2) Fuel economy projections for future gas turbines have been made by several groups and the results span a considerable range. The JPL fuel economy projections including the influence of size-effects on component efficiencies, are discussed in Section 9.
- (3) Available information on combustors indicates that future gas turbine-powered cars should be able to meet the research goal emission standards of 0.4 g/mi HC, 3.4 g/mi CO, and 0.4 g/mi NO_x. The only possible

difficulty envisioned is that of meeting of 0.4 g/mi NO_x standard for the ceramic engine with a TIT = 2500°F. The multifuel capability of the automotive gas turbine has been confirmed in numerous vehicle tests.

3.7.3 Critical Problems

A number of critical problems have been identified which must be dealt with successfully before the gas turbine engine is likely to yield fuel economy attractive to the automotive industry:

- (1) Selection of engine configurations best suited for passenger car applications.
- (2) Development of variable-geometry components which are practical and cost-effective.
- (3) Development of ceramic materials and components.
- (4) Minimization of size-effects on component efficiencies.
- (5) Optimum integration of the engine and transmission into a powertrain unit.

3.7.4 Recommended R&D Areas

As would be expected, the current DOE Gas Turbine Program has R&D activity in most of the areas identified as being critical to future gas turbine engine development. There are, however, a number of areas in which it is recommended that new work be initiated and/or present activity be expanded:

- (1) Modular engine design and testing approaches to be used to more expeditiously evaluate alternative component designs.
- (2) Increased activity in the development and evaluation of variable geometry components.
- (3) Experimental verification of size-effects on component and engine efficiencies for 50-100 hp engines and high turbine inlet temperatures (up to 2500°F).
- (4) Development of a low NO_x emissions combustor for use in a ceramic engine (TIT - 2500°F).
- (5) Evaluation of the trade-offs between engine packaging, efficiency, and cost and vehicle weight, performance, and fuel economy.

- (6) Study of the use of transmission/energy storage combinations to load-level both two-shaft and single-shaft engines.
- (7) Development of engine/transmission/vehicle simulation models and computer programs which realistically account for the effect of transients on vehicle performance and fuel economy.

3.8 ADDENDUM: EFFECTS OF SMALL ENGINE SIZE - ANALYSIS OF GAS TURBINE ENGINES OF SMALL POWER OUTPUT AND HIGH TURBINE INLET TEMPERATURE

3.8.1 Introduction

A still unresolved problem associated with the development of gas turbines for passenger cars is degraded efficiency with reduced engine size. For example, as shown in Figure 3-63, experience with the development of gas turbines shows that component efficiencies decline at flow rates of less than 2 kg/sec (4 lb/sec), which for metal engines is a power output of about 300 hp. For engines of 100 hp or less the efficiencies that can be achieved drop even more sharply unless special care such as rework and finishing touches, selective assembly, etc. is rendered the components.

The basic reasons for the degradation of efficiency with reduced size are known. For example, the relationship between viscous losses, Reynolds number, and relative surface roughness is shown, for pipe flow resistance, in Figure 3-64, and in terms of compressor quality in Figure 3-65. This effect is primarily responsible for the initial size degradation which, as mentioned above, becomes noticeable at about 300 hp and below. The growth of the viscous losses with decreasing Reynolds number is relatively flat for a hydraulically smooth surface condition such as that resulting from state-of-the-art manufacturing precision and surface finishing. Fabricating components in that way is feasible only for research hardware. With actual production hardware the loss-size relationship is strongly governed by relative surface roughness, which tends to grow as engine size decreases. The compounded (and realistic) loss-size relation indicated in Figure 3-65 is stronger than the Reynolds number-loss relation for a given level of surface roughness. At critical or near-critical expansion velocities, such as usually prevail in turbine nozzles, Reynolds numbers are high and their effect on losses is relatively weak. In that case, size degradation such as is shown in Figure 3-66B is primarily governed by the relative growth of surface roughness.

The stronger degradation of turbine and compressor efficiencies in engines of 100 hp or less is due to the compounding of losses introduced by manufacturing technology limitations such as hardware tolerances, minimum producible trailing edge thickness of turbine and compressor blades, and by contour imperfections such as those shown in Figure 3-67. From a certain size and smaller, which seems to be about

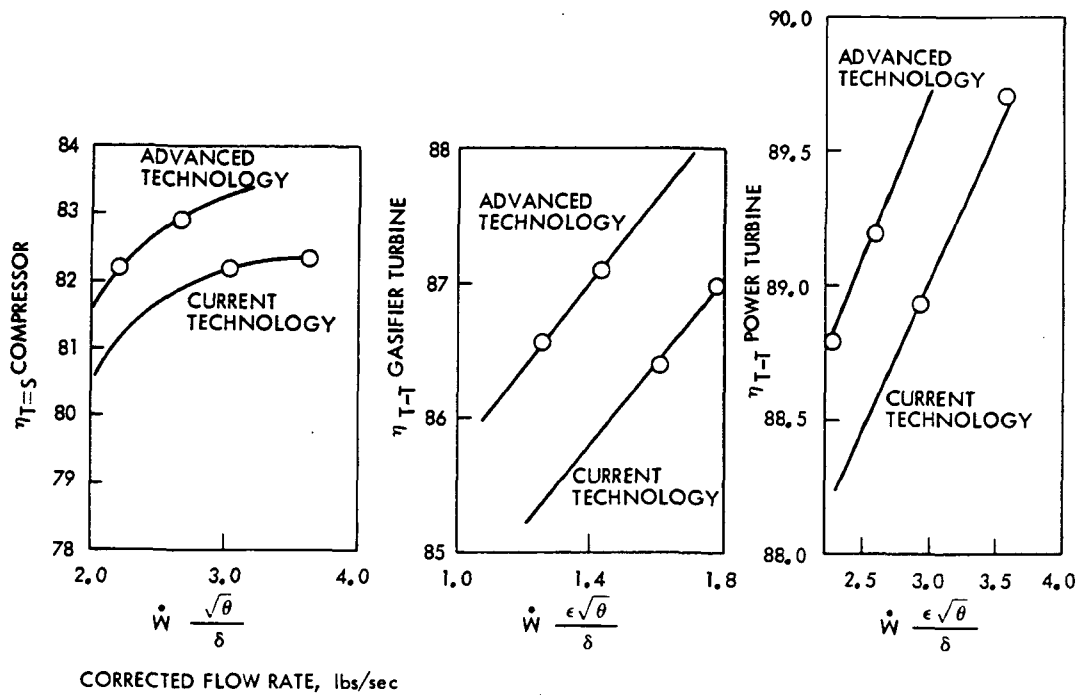


Figure 3-63. Effects of Size on Component Efficiency of Production Gas Turbine Engines (Reference 3-40)

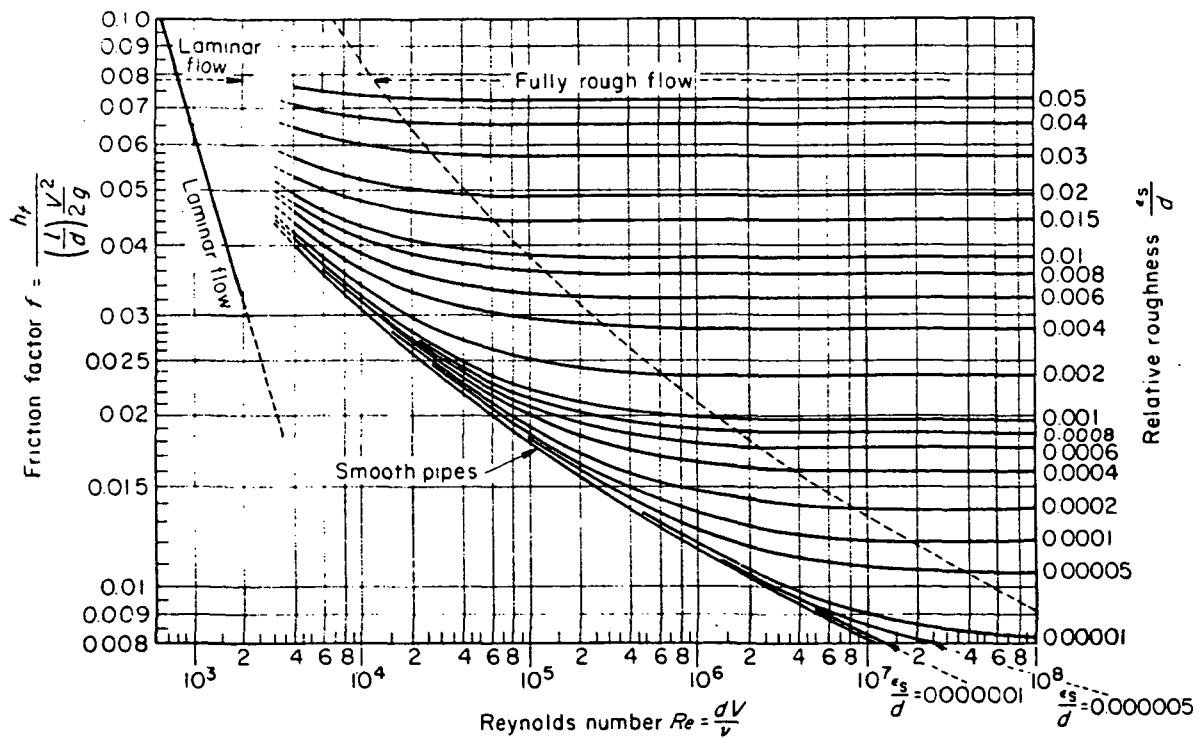
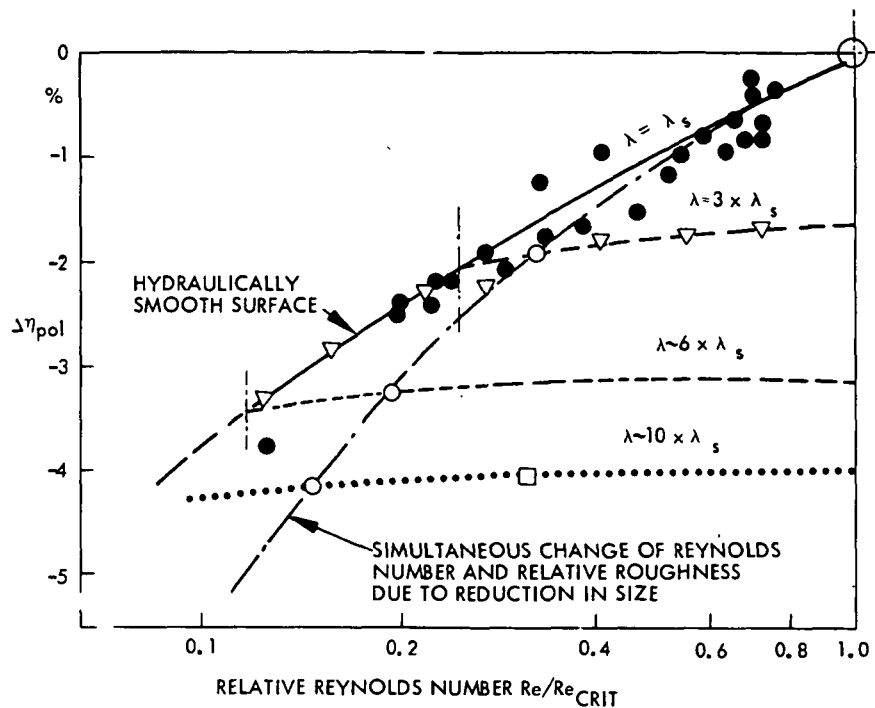


Figure 3-64. Effect of Reynolds Number and Surface Roughness on Flow Resistance Through Pipes (Reference 3-41)



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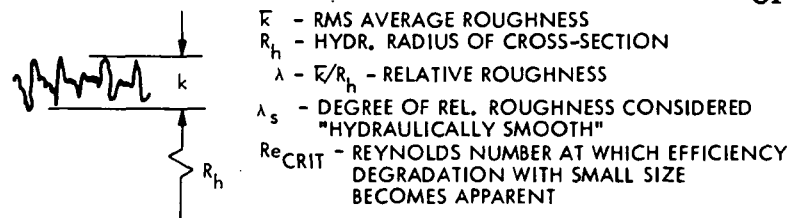
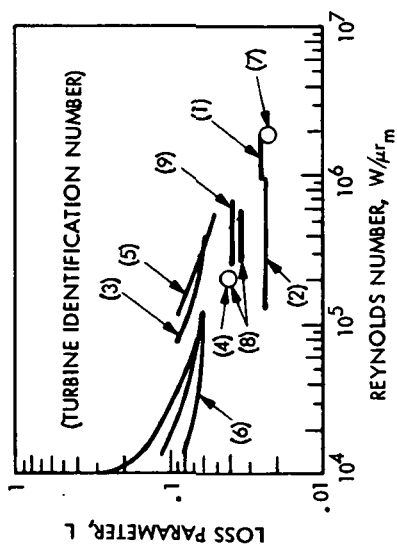


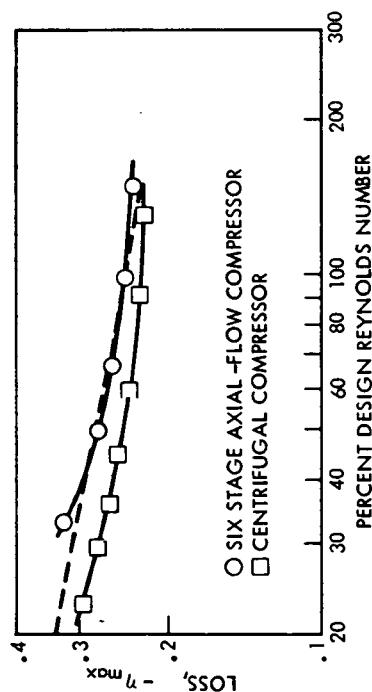
Figure 3-65. Typical Relations between Efficiency Degradations, Relative Reynolds Number and Surface Roughness (Reference 3-42)

100 hp with present production technology, a proportionate scaling of hardware dimensions is not possible. This causes the form losses due to trailing edge blocking and wake mixing to increase with decreasing size after loss-critical dimensions are reached. The sudden increase of the losses shown in Figure 3-66B indicates that a relative growth of loss-critical dimensions takes place for throat areas below 1 cm². Where fractional running clearances cannot be maintained with downscaling, efficiencies will be impaired with decreasing size according to the rates shown in Figure 3-68.

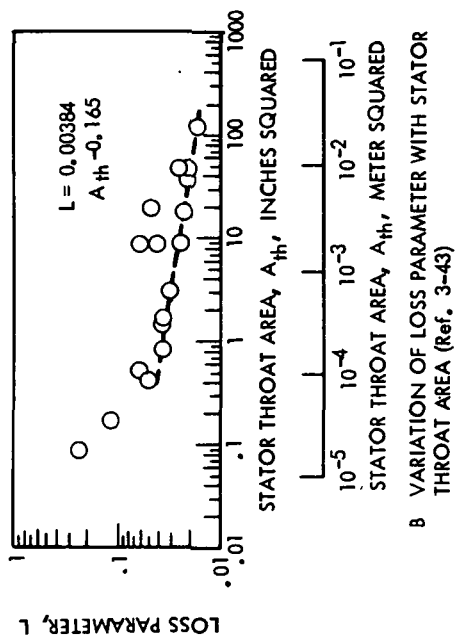
The above physical reasons for size degradation, in particular the problems associated with Reynolds number, surface roughness, and radial clearance, have been addressed in a variety of researches. Unfortunately, most work done in these areas deals only with one problem at a time without controlling the others, and the data obtained are frequently the result of unknown compounded effects. Being also



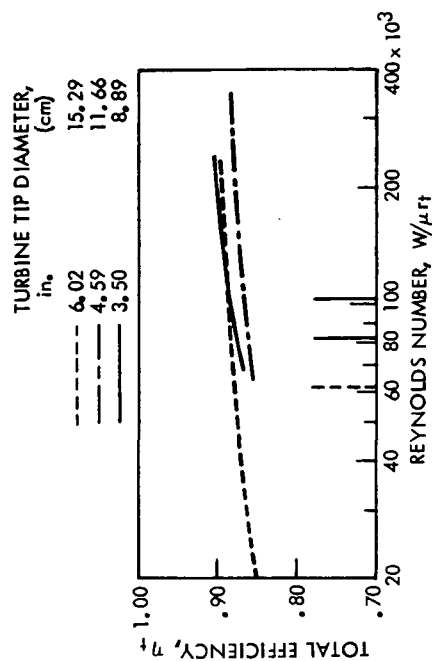
A VARIATION OF LOSS PARAMETER WITH REYNOLDS NUMBER (Ref. 3-43)



C LOSS AS FUNCTION OF PERCENT DESIGN REYNOLDS NUMBER AT DESIGN SPEED (Ref. 3-44, 3-45)



B VARIATION OF LOSS PARAMETER WITH STATOR THROAT AREA (Ref. 3-43)



D COMPARISON OF EFFICIENCY AS A FUNCTION OF REYNOLDS NUMBER BETWEEN THREE RADIAL INFLOW TURBINES OF DIFFERING TIP DIAMETER (Ref. 3-44, 3-45)

Figure 3-66. Results of Recent Research with Small Turbines and Compressors Conducted by NASA

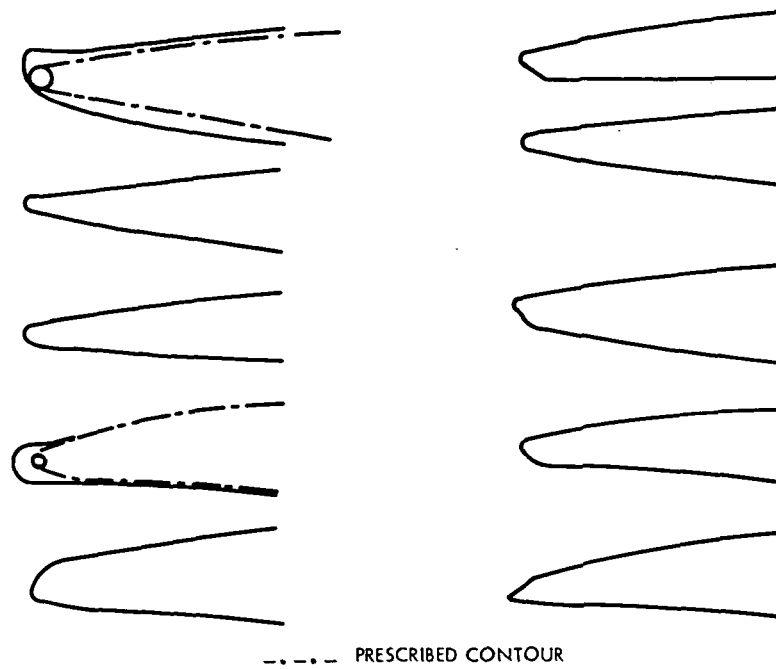


Figure 3-67. Typical Contour Deficiencies Encountered on Leading Edges of Mass-Produced Small Turbine and Compressor Blades (11 to 16 mm Chord) (Reference 3-42)

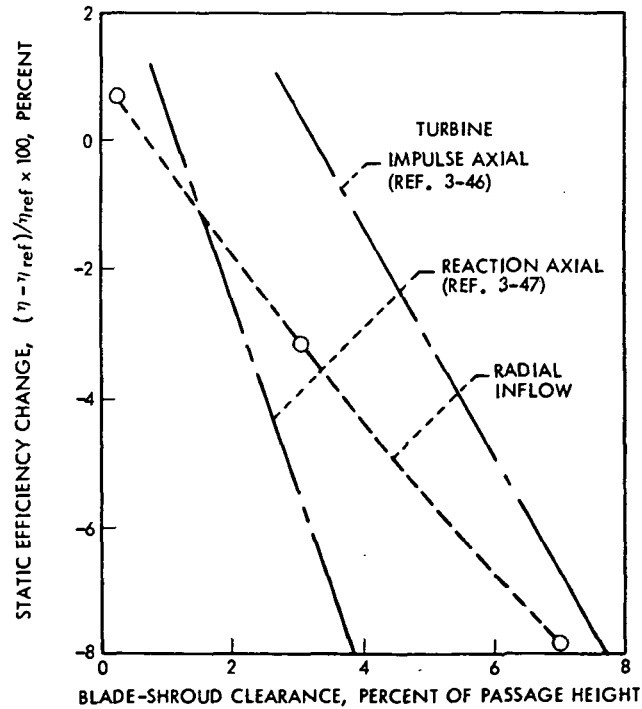


Figure 3-68. Effect on Static Turbine Efficiency of Clearance Variation (Reference 3-48)

dependent on initial turbulence and compressibility effects, the approach used to vary the Reynolds number has a strong effect. For example, changing velocity alone introduces a change of Mach number effect unless incompressible flow relations are applicable. Changing size at constant velocity means changing relative roughness and surface quality. For compressible flow, only density changes at constant Mach-number truly reflect the effects of Reynolds number alone. Unfortunately, most tests conducted with turbo-machinery components represent a combination of variables, which makes it difficult to compare the results obtained from different investigators and to draw general conclusions. Most researchers agree that the Reynolds-number-loss-relations for turbines and compressors can be expressed in the form

$$\xi/\xi_{crit} = a + b (Re/Re_{crit})^m$$

where ξ represents a loss coefficient, and Re a Reynolds number as defined by the investigator. The subscript "critical" denotes those conditions below which Reynolds number effects are noticeably growing. This has nothing in common with the "critical Reynolds number" that defines the transition from laminar to a turbulent boundary layer flow. As Table 3-23 shows, the constants a , b , and the exponent m derived by different investigators vary widely. The above-discussed compounding of physical effects not related to the Reynolds number, as well as differences associated with free stream turbulence, the test setup, and measuring techniques are responsible for this.

Table 3-23. Values for a , b , and m as Established by Various Investigators (Reference 3-49)

Researcher	Constants			Definition of Reynolds Number Used
	a	b	m	
Ackeret	0.5	0.5	0.2	$D_2 u_2 / v_0$, where D_2 = impeller outlet diameter
Pfleiderer	0	1	0.1	
Hutton	0.3	0.7	0.2	
Wiesner	0.5	0.5	0.1	
Fauconnet	0.24	0.76	0.2	
Davis	0	1	0	
Carter	0	1	0.2	$L_1 c_1 / v_1$, where L_1 = impeller channel length
Bullock	0	1	0.2	

The NASA-Lewis Research Center has conducted numerous tests with small turbines and compressors with the objective of establishing the characteristics of small solar or other heat source-powered closed-cycle Brayton engines at low Reynolds numbers. Some of the results published are shown in Figure 3-66. Although reflecting the capability of state-of-the-art research hardware, these tests have been cited as the basis for the assessment of the performance of gas turbines applicable to small and medium-sized cars and have led to the conclusion by some workers that small-size penalties are not a significant problem. Unfortunately, experiences to date with automotive gas turbines, including truck engines and in all likelihood the Chrysler Upgraded engine as well, do not seem to confirm that conclusion.

3.8.2 Analytical Approach

This section discusses a simple analytical approach to assessing the compound effects of power, temperature, and size on engine losses relative to a near-size-critical engine⁽¹⁾ that develops 100 hp at TIT = 1925°F. It is assumed, optimistically, that the quality and precision achievable to date on research hardware will be possible with automated production methods in the future. The thermodynamic model used to determine flow-rates, engine size, and specific fuel consumption is shown schematically in Figures 3-69, 3-70, and 3-71. Allowance was made for the change of the thermodynamic parameters with temperature according to Keenan and Keye (Reference 3-50). The characteristics of

(1) By definition, the engine size below which a proportionate downscaling of loss-critical dimensions is not feasible.

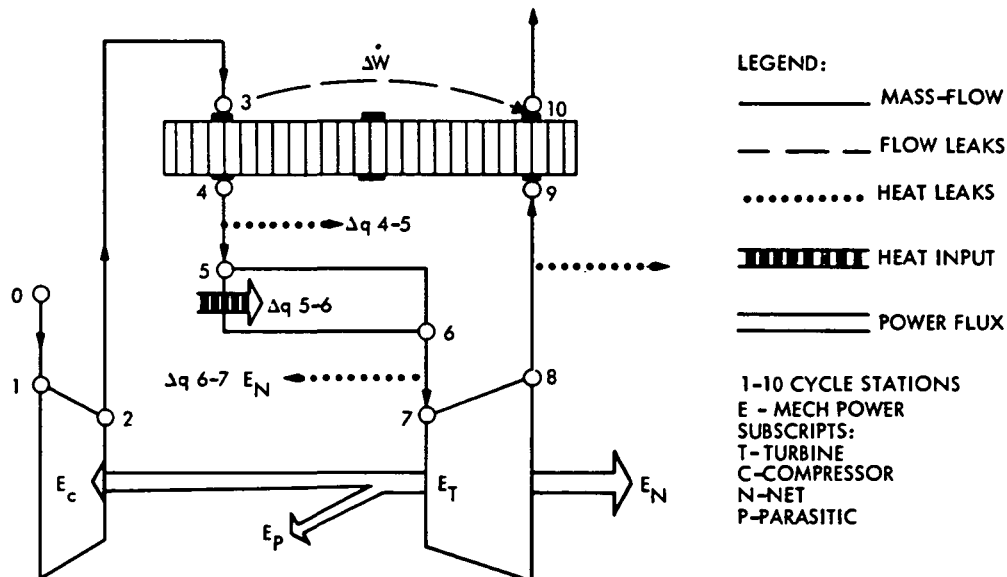
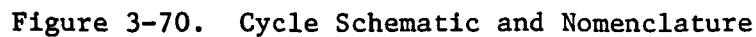


Figure 3-69. Engine Schematic, Flow Diagram, and Nomenclature



With turbine inlet temperature and engine power output being the primary variables, size-related effects on major engine components and their efficiencies were determined in an iterative manner, taking incremental steps from the reference engine toward lower power and higher turbine temperatures, until convergence between size and related losses was reached. All computations were performed for a constant pressure ratio of 4 and for constant pressure losses between the compressor and the turbine as indicated in Table 3-24.

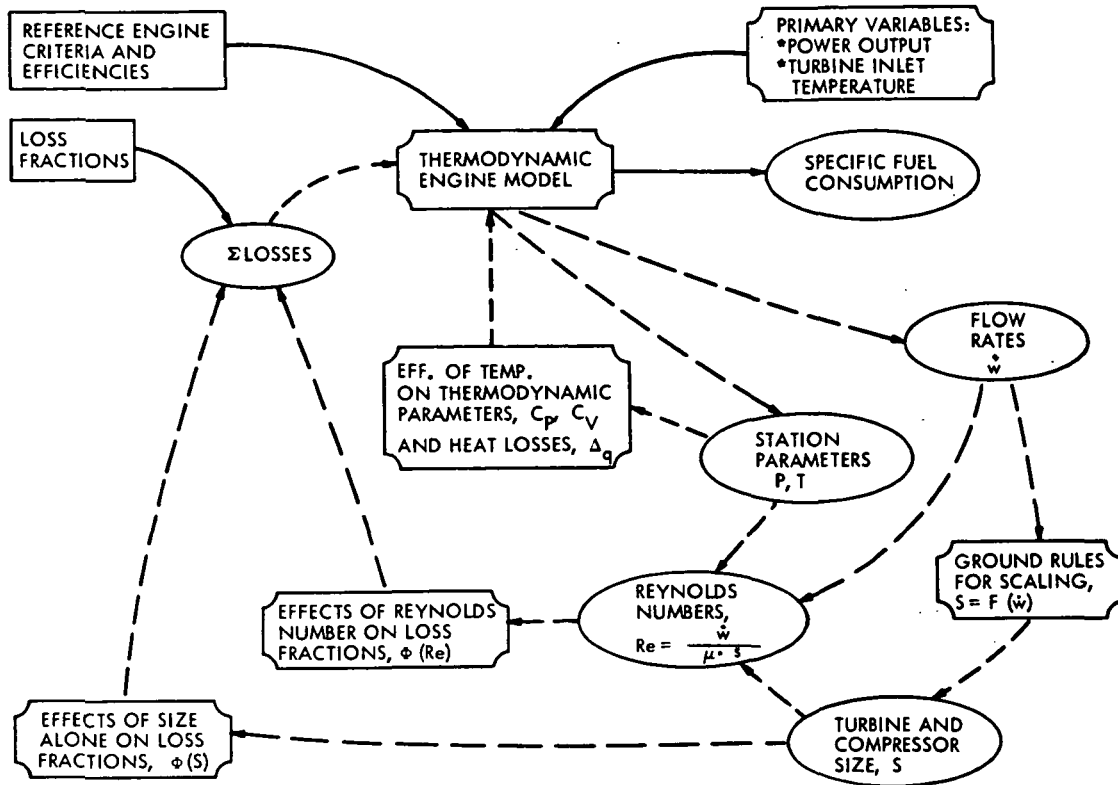


Figure 3-71. Logic Flow for the Iterative Determination of Engine Losses Including Size Effects

For the relationship between flow rate and size it was assumed that the Mach numbers were constant and square scaling relations held. Hence,

$$\dot{W} T^{1/2} / p S^2 = \text{const.} \quad (1)$$

where $\dot{W}$ is the flow rate, S a characteristic dimension, and p and T are the absolute pressure and temperature at the compressor or the turbine inlet. For the compressor, p and T remain constant, and the general size-flow rate relationship becomes truly square, i.e.

$$(S/S_{\text{crit}})_{\text{comp}} = (\dot{W}/\dot{W}_{\text{crit}})^{1/2} \quad (2)$$

Table 3-24. Design Parameters of 100 hp Reference Engine
(Reference 3-51)

Parameter	Value	Dimension
Compression Ratio	4.0	--
Turbine Inlet Temperature	1925	°F
Power Concentration (unaugmented)	77.6	hp/lb
Specific Fuel Consumption	0.440	lb/hp-h
Efficiencies:		
Compressor	0.785	--
Combustor	0.990	--
Turbine	0.840	--
Heat Recovery	0.900	--
Mechanical (including parasitic losses)	0.900	--
Pressure Drop:		
Inlet	0.015	--
Compressor to Turbine	0.024	--
Turbine to Atm.	0.090	--
Heat Losses:		
Regenerator to Combustor	0.680	Btu/lb
Combustor to Turbine	0.110	Btu/lb
Turbine to Regenerator	0.230	Btu/lb
Regenerator Leakage	0.045	--

For the turbine, the inlet pressure remains constant and the inlet temperature is varied. Using Eqn (1) the flow-size relation for the turbine becomes

$$(S/S_{crit})_{turb} = (T/T_{crit})^{1/4} (\dot{W}/\dot{W}_{crit})^{1/2} \quad (3)$$

Define the Reynolds number as

$$Re = W/\mu S \quad (4)$$

where μ is the flow viscosity. For the compressor, the inlet temperature T and consequently the viscosity are constant, and Reynolds numbers change in the same ratio as size, i.e.

$$(Re/Re_{crit})_{comp} = (S/S_{crit})_{comp} = (\dot{W}/\dot{W}_{crit})^{1/2} \quad (5)$$

According to Sutherland (Reference 3-52) the effect of temperature on viscosity can be expressed in the form

$$\mu/\mu_{crit} = (1 + C/T_0)(T/T_0)^{1/2}/(1 + C/T)$$

and, for high temperature, with good approximation in the case discussed, by the relationship

$$\mu/\mu_{crit} = (T/T_{crit})^{1/2} \quad (6)$$

Substituting Eqns (6) and (3) in Eqn (4), the Reynolds number change for the turbine

$$(Re/Re_{crit})_{turb} = (\dot{W}/\dot{W}_{crit})^{1/2} (T/T_{crit})^{-3/4} \quad (7)$$

The determination of the turbine flow rate makes allowance for regenerator leakage according to

$$W_T = W_c - \Delta W = W_c (1 - \Delta \dot{W}/\dot{W})$$

where $\Delta \dot{W}/\dot{W}$ represents the size-dependent fraction of the flow bypassing the turbine. Since the fuel air ratio is high (~ 100) in regenerative engines, the addition of the fuel to the mass-flow rate is neglected.

The general relationships and the constants used to assess the effect of Reynolds number and size on typical loss-fractions are listed in Tables 3-25 and 3-26. It is conservatively assumed that the effects of size on viscous losses (ξ_v) are more pronounced in the compressor (decelerated flow, $m = 1/2$) than in the turbine ($m = 1/5$) and that secondary flow losses change according to Figure 3-72. It is assumed that the absolute rotor clearances remain constant at engine sizes below 100 hp. Thus, based on the relationship represented by the

Table 3-25. Effects of Reynolds Number and Size on Loss Fractions In Terms of Ratio Relative to Size-Critical Engines

Type of Loss	$\phi/\phi_{crit} = F(x)$	x
Viscous	$a + b (x)^{-m}$	Re/Re_{crit}
Secondary		
Trailing Edge Blockage	$K_{mx} (x)^{-\alpha}$	S/S_{crit}
Rotor Clearance		

Table 3-26. Summary of Constants Used in Size-Effects Analysis

Constant	Compressor	Turbine	Regenerator	Remarks
a	0 ⁽¹⁾	0 ⁽²⁾		(1) Ref. 3-49
b	1	1		(2) Ref. 3-53
m	0.5	0.2	--	
n	0.5			
k _{mx}	1 ⁽³⁾	1 ⁽⁴⁾		
α	2	2		(3) Ref. 3-54
k _{cl}	1	1	--	(4) Ref. 3-53
β	1	1		
k _{lk}	--	--	1	Assumes constant leak rate per unit length
γ			1	

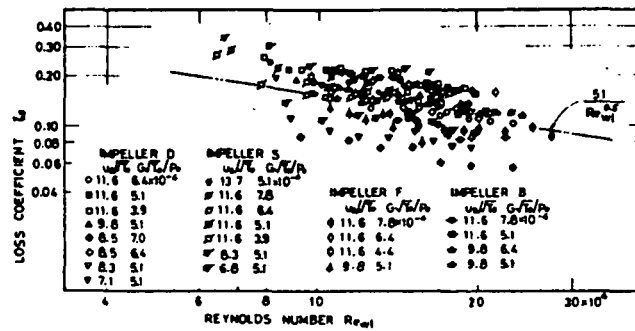


Figure 3-72. Relationship Between Secondary Flow Loss Coefficient and Impeller Channel (Re_{w1}) Reynolds Number (Reference 3-54)

dashed curve in Figure 3-68, clearance losses will change in inverse proportion to size. The square relationship between wake-mixing losses and size results from the assumption that they are represented with good approximation by the losses that theoretically would result from the sudden expansion due to trailing edge blockage according to $\xi_{ms} = 2g (\Delta V)^2/V^2$ where V is the expansion velocity, and ΔV is the velocity change that takes place after complete mixing. For the relationship between regenerator leakage and size, it is assumed that the flow leaking across the seals is critical (i.e., choked) and the rms clearance between the core and the seal remains constant according to a practical minimum. The leak rate per unit seal length is then a constant in absolute terms, and the relative leakage changes with

$$\frac{\Delta \dot{W}/\dot{W}}{(\Delta \dot{W}/\dot{W})_{crit}} = (S/S_{crit})^{-1} \quad (8)$$

The various loss fractions for the size-critical reference engine are listed in Table 3-27. The compressor and turbine efficiencies of all the other engine designs ($W < W_{crit}$, and $T > T_{crit}$) can then be determined from the equation

$$(1 - \eta)/(1 - \eta_{crit}) = \Sigma \phi_{crit} \cdot F(X) \quad (9)$$

by inserting the X according to Eqns (2), (3), and (7).

Allowance was made in the thermodynamic cycle calculations for changes in heat losses across housing walls in the locations

Table 3-27. Summary of Loss Fractions Derived for
100 hp Reference Engine

Type of Loss	Loss Fractions $\phi = \zeta \times / \zeta_{\text{tot}}$	
	Compressor	Turbine
Viscous	0.322	0.371
Secondary	0.315	
Wake Mixing	0.061	0.184
Clearance	0.091	0.085
Remaining	0.211	0.360
Total	1.000	1.000
Remarks	Derived from Ref. 3-54	Derived from Refs. 3-48, 3-53, 3-57

indicated in Figure 3-69, assuming that the heat flux per unit surface area across the insulations is proportional to the difference between the total gas and ambient temperatures, i.e.,

$$\Delta q / \Delta q_{\text{crit}} = (T - T_a) / (T_{\text{crit}} - T_a) \quad (10)$$

where T_a is the ambient temperature.

3.8.3 Results

Results obtained to date using the model described in the previous section are given in Figures 3-73 to 3-76. The figures show the effect of design power and turbine inlet temperature on engine flow rate, the size and the Reynolds numbers of the compressor and the turbine in relation to the 100 hp reference engine, and the change of major component efficiencies. It can be seen that the changes effected by turbine inlet temperature are relatively flat whereas the changes associated with power and size are progressive. The X^n -type relations between flow rate, size, and Reynolds number are primarily responsible for this.

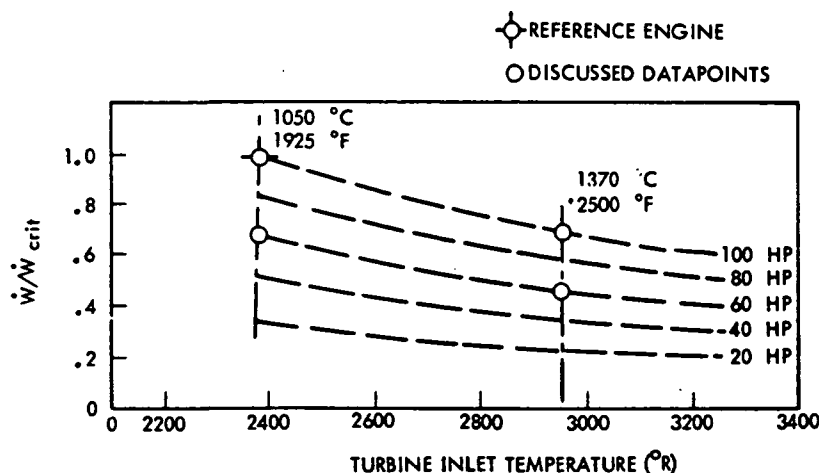


Figure 3-73. Effect of Turbine Inlet Temperature and Power Output on Engine Flow Rate Relative to Reference Engine (100 hp/1925°F)

The compound effects of power and turbine temperature become apparent from Figure 3-77. A reduction in power from 100 to 60 hp would, for example, result in an 8 percent increase in the specific fuel consumption if the TIT remains at 1925°F (1050°C), and an increase of 18 percent if the design power were reduced further to 40 hp. The turbine temperature would have to be raised to 2200°F (1200°C) just to compensate for the size penalty for a 60 hp engine, and to 2500°F (1370°C) for a 40 hp engine.

In case of the 100 hp engine, size-related penalties (represented by the cross-hatched area in Figure 3-77) are introduced with higher turbine temperatures due to increased power concentration and consequently reduced engine size and Reynolds numbers. The improvement that would be seen without size-penalties for a turbine temperature increase from 1900°F to 2500°F is approximately 25 percent greater than that which is calculated when size penalties are accounted for.

The assumptions made and the input data chosen for this study are subject to disagreement. It should be born in mind, however, that the calculated trends and the order of magnitude of size-effects were the primary objectives of the analysis. The results obtained, however, seem to be in reasonable agreement with calculations conducted by others, such as the Chrysler prediction (Reference 3-55) indicated by the solid triangle plotted in Figure 3-77.

3.8.4 Design Approaches to Reduce Size Effects

The study has concluded, in essence, that engine performance and fuel economy degradation are progressively larger at powers below 100 hp to a degree that makes the viability of gas turbines for small

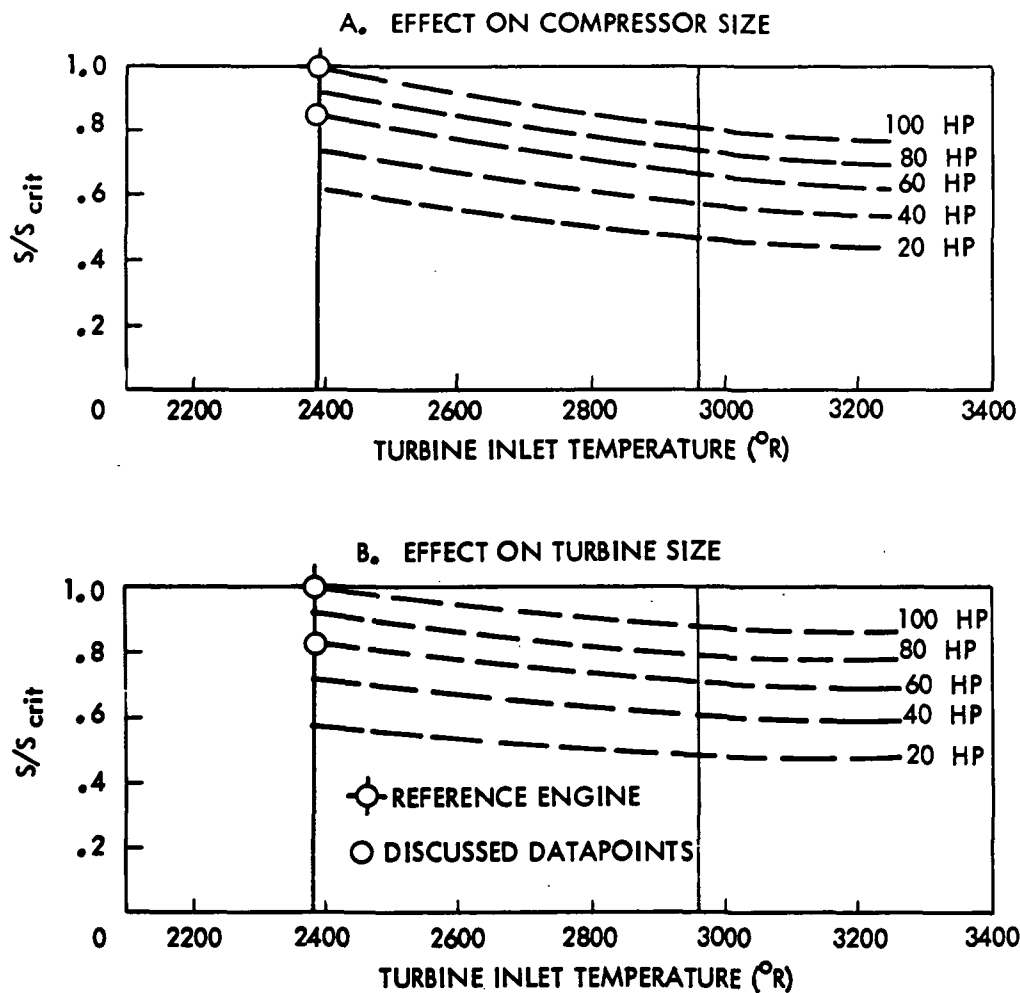


Figure 3-74. Effect of Turbine Inlet Temperature and Power on Size of Compressor and Turbine Relative to Reference Engine (100 hp, 1925°F)

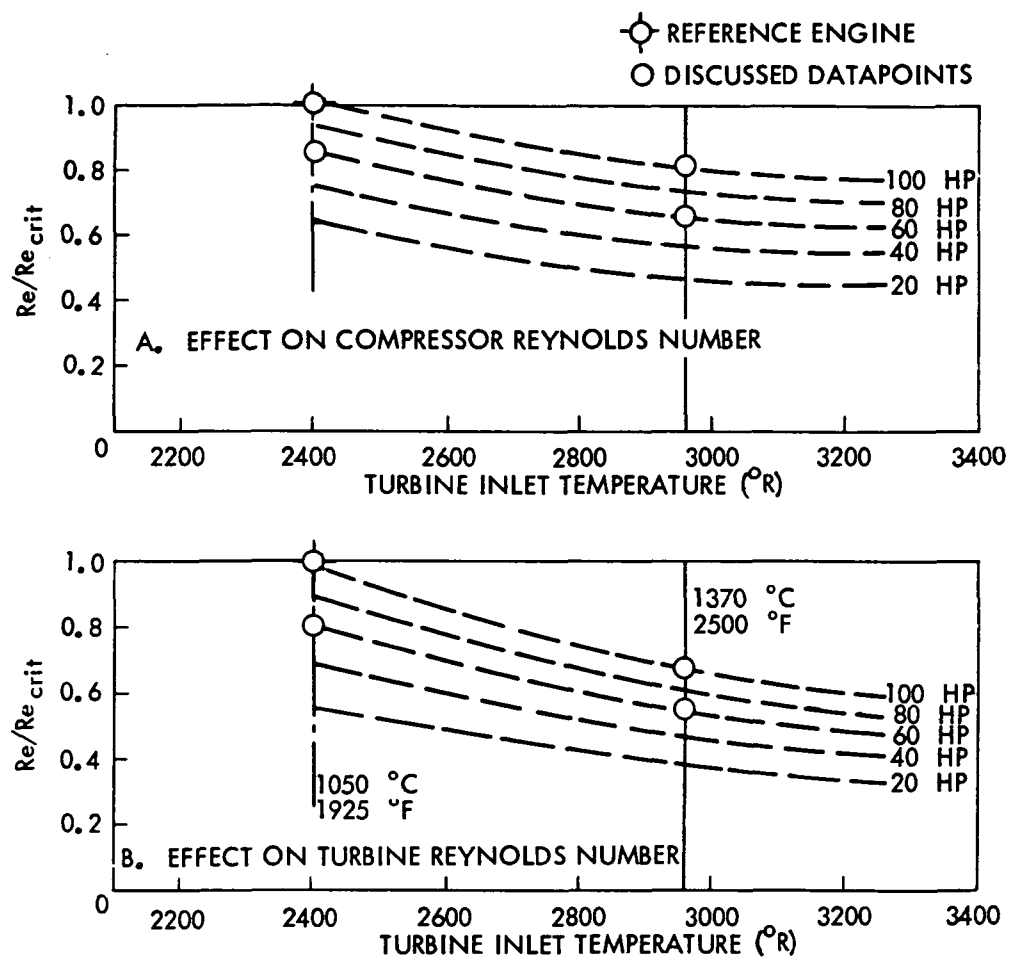


Figure 3-75. Effect of Turbine Inlet Temperature and Power on Compressor and Turbine Reynolds Numbers Relative to Reference Engine (100 hp, 1925°F)

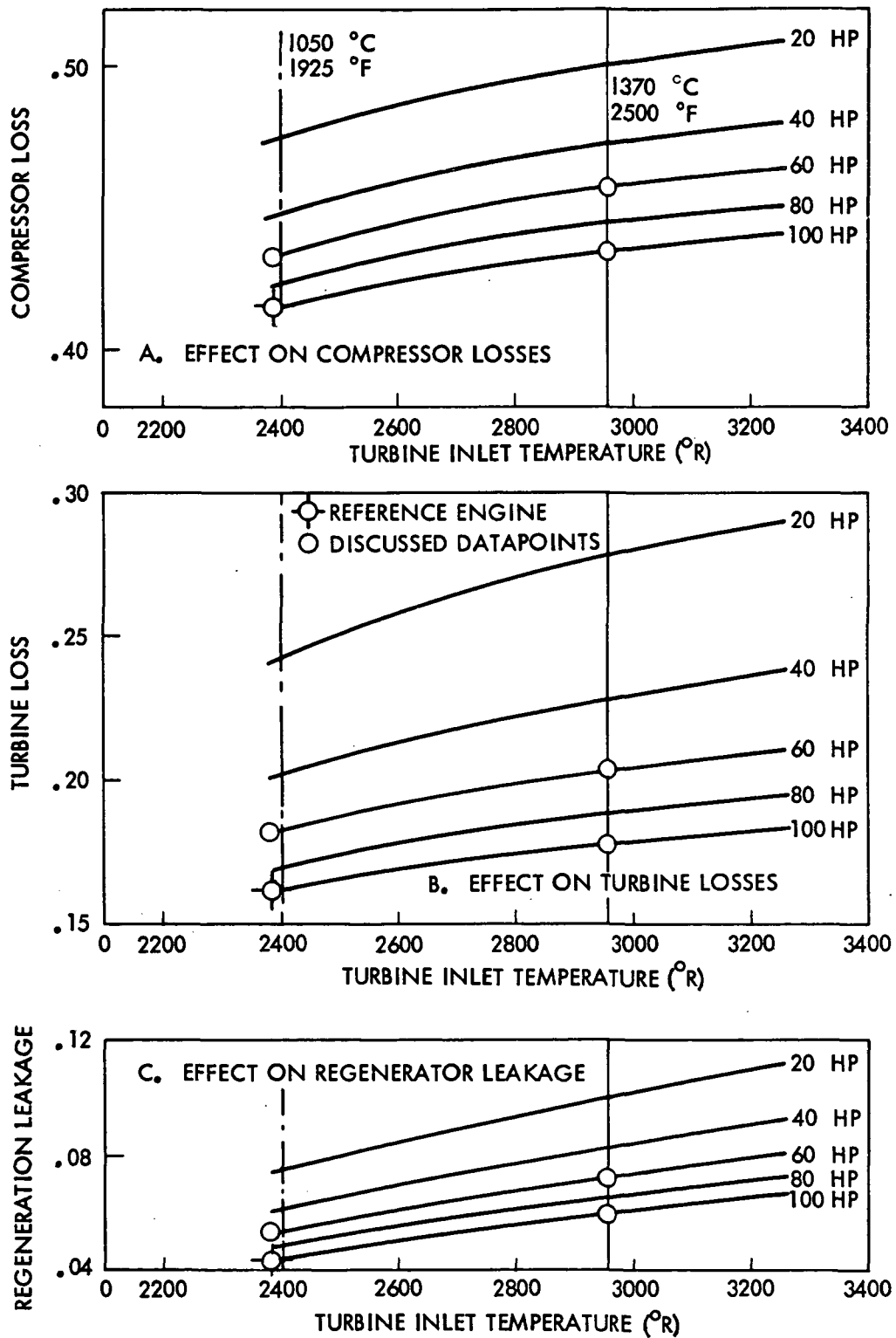


Figure 3-76. Effect of Turbine Inlet Temperature and Power on Change of Engine Losses

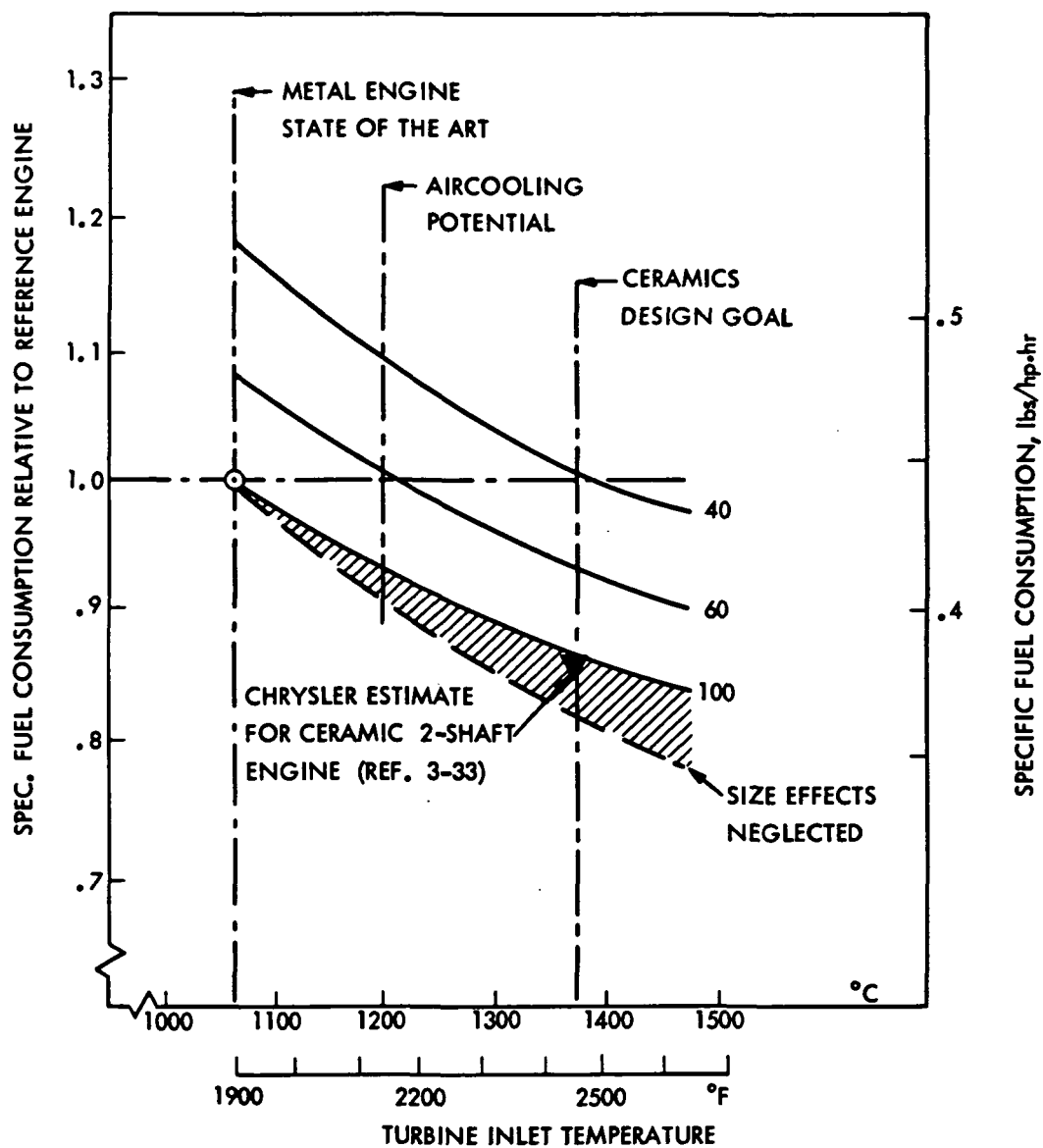


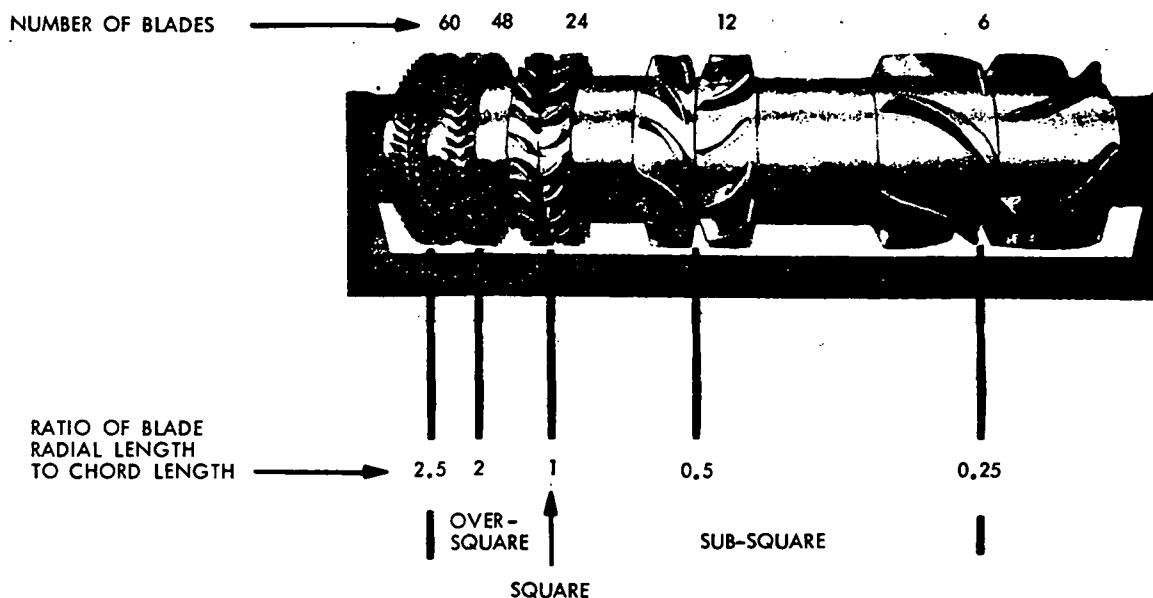
Figure 3-77. Effect of Turbine Inlet Temperature and Design Power on Engine Specific Fuel Consumption

cars questionable. The general degradation of aerodynamic quality due to relative growth of surface roughness and imperfections, hardware tolerances, and minimum wall thicknesses seems to be the primary contributor to size-penalties over Reynolds number effects, which are frequently considered to be the primary cause for the efficiency degradation due to size.

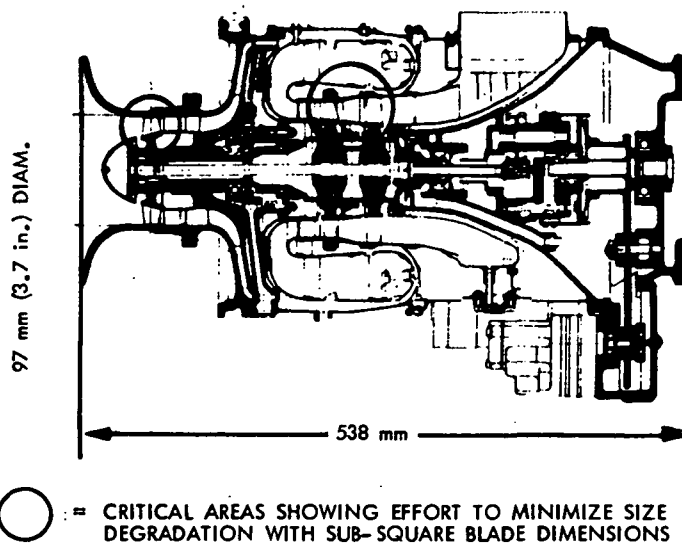
A smaller radial clearance than assumed can probably be provided in a ceramic turbine because of the high modulus of elasticity and the low thermal expansion of the ceramic candidate materials. On the other hand it is expected that, dictated by ceramic production and design technology, the aerodynamic quality degradation of turbine blades and trailing edge blockage with size will be more pronounced than for metal components. It is a future objective of this continuing study to establish the sensitivity of size-effected losses to a variation of the design and manufacturing related parameters.

Possible ways of reducing size effects were also considered briefly. Approaches for the improvement of small turbines and compressors generally require an increase of the hydraulic diameters of size-critical passages such as the cross-sectional flow areas of diffusers and nozzles and the areas between impeller and turbine blades. To accomplish this requires a reduction in the number of blades and increased blade length to maintain solidity. This will increase the Reynolds number, reduce relative roughness, and in general improve aerodynamic quality, although secondary flow losses would tend to increase. Square or even sub-square turbine blade dimensions where needed are widely considered as the best aerodynamical compromise for small turbines and compressors. As shown in Figure 3-78, a disadvantage of this approach is that wider and heavier discs are required and the moment of inertia of the rotor is relatively high. Since, as mentioned earlier, inertial effects during urban vehicle operations have a significant effect on vehicle fuel economy, approaches which increase engine inertia may not be appropriate for automotive gas turbines. Conical rather than rectangular diffusion channels seem to be more advantageous from the size-efficiency standpoint.

Since engine size changes with the square root of the flow velocity at a given flow rate, a general reduction of flow velocities through the engine should be an effective approach to combating the size problem where trailing edge blocking and general aerodynamic quality rather than Reynolds number are the primary contributors. Lower flow velocities mean reduced Mach-number effects and stresses, deeper blades (or vanes), more favorable hydraulic diameters of size-critical flow passages, relatively larger nose radii, and reduced sensitivity to angle-of-attack changes (i.e., in essence a more forgiving engine behavior in regard to small-size effects and part-load and transient operation). The implementation of lower flow velocity leads to multispool concepts or at least to a two-stage compressor. Although mechanically more complex and expensive, multispool designs are attractive from the mass-production and reliability standpoints. At the low compression ratios required for regenerative engines (4 to 5), this will allow for the utilization of lower (turbosupercharger) grade components, which are



A. Compressor Rotor-Stator Cascades of Various Sizes With Equal Solidity, Flow Rate, and Performance (Ref. 3-56)



B. Kloeckner Humboldt Deutz 114 kW Gas Turbine T 312 (KHD drawing)

Figure 3-78. Reducing Size Degradation by Lowering Blade Aspect Ratios

relatively inexpensive compared to the highly stressed rotary components of high-performance single-stage compressors and turbines. Inertial effects should also be less pronounced with lower stress levels, lighter discs, and lower shaft speeds.

A problem usually not mentioned in conjunction with small single-shaft engines is that of power takeoff and reduction gearing. Since the performance of aerodynamic compressors and turbines depends upon circumferential speed in absolute terms, shaft velocities increase proportionately with decreasing size. This will lead eventually to a situation at which a mechanical power takeoff from the rotor becomes critical or not feasible from the gear design standpoint. Approaches to a gas-turbine integrated electrical power takeoff could be required for small automotive power plants if gearing becomes the size-limiting factor.

SECTION 4

STIRLING ENGINES

4.1 INTRODUCTION

A Stirling engine is a closed cycle machine which converts thermal energy to mechanical work by alternately compressing and expanding a confined working fluid with heat rejection during compression and heat addition during expansion. The cyclic movement of the working fluid in the engine is accomplished by volume changes in the working volume rather than intermittently closed valves. The cycle includes thermal regeneration which is accomplished with a storage heat exchanger that removes heat from the working fluid after expansion and returns heat to the working fluid after compression.

Interest in the Stirling engine for automobile application stems primarily from its potential for good fuel economy and low exhaust emissions. Theoretically, the thermal efficiency of the ideal Stirling cycle is equal to that of a Carnot cycle engine, whose efficiency is the maximum attainable for a heat engine operating between two temperature limits. Although difficult to approach in actual hardware, this potential of the Stirling engine for high thermal efficiency has become an increasingly important factor because of its potential effect on future National energy needs. Although many different heat sources could be used, the Stirling engines being considered for automobile application use a continuous, low-pressure combustor as the heat source. Properly designed combustors of this type can meet the most stringent anticipated environmental standards for emissions of hydrocarbons, carbon monoxide, and nitrogen oxides without a penalty in fuel consumption.

In addition to good fuel economy and low emissions characteristics, the Stirling engine has other desirable attributes for automobile applications. The continuous, low-pressure combustor has the capability, with minor modifications, of using a broad variety of liquid fuels, gaseous fuels, and nonpetroleum fuels which may be produced in coming years. The combustor is insensitive to the detailed fuel composition and variations in such fuel characteristics as octane number, volatility, and viscosity. As petroleum becomes less plentiful, this multifuel capacity become increasingly important. The Stirling engine also has a very low noise level. The continuous, low-pressure combustor is inherently quieter than the conventional gasoline or diesel engines with their high-pressure, intermittent combustion process.

The first engine operating on the Stirling cycle was devised by Robert Stirling in 1816. Early development of the engine placed it in competition with the steam engine as the power source for ship propulsion and water pumping. These Stirling engines used air as the working fluid. The steam engine prevailed because of its superior power/weight characteristics.

Since good historical reviews of Stirling engine development are available in References 4-1, 4-2, and 4-3, only some of the more important events related to the automotive application will be mentioned here. Most of the automotive-related developments of the Stirling engine have involved N. V. Philips Gloeilampenfabrieken of The Netherlands, either directly or through various licensing agreements. Historically, Philips began a research and development effort in Stirling engines in 1938 in their search for a portable power generation unit for radios and other small electrical equipment. From 1958 to 1968, General Motors had a licensing agreement with Philips. Research was conducted in many fields where the application of Stirling engines looked attractive. General Motors Research Laboratories constructed a Stirling engine-electric hybrid car to evaluate the use of this low-emission engine and electric drive system in a small car (Reference 4-4). In 1968, Philips signed licensing agreements with Maschinenfabrik Augsburg-Nuremberg, jointly with Motoren-Werke Mannheim (MAN-MWM), in Germany and with United Stirling of Sweden (USS). Some of the Stirling engine activities of MAN-MWM are given in Reference 4-5.

In 1971, Philips and Ford Motor Company began a joint effort to evaluate the feasibility of a Stirling engine for automotive propulsion. As a result, Philips signed a license with Ford Motor Company in 1972 for the joint development of the 4-215 Stirling engine. A Philips/Ford 4-215 (170 bhp) engine was installed in a Ford Torino (4500 lb inertia weight) for initial vehicle testing and evaluation in 1975. That same year, Ford was funded by the Energy Research and Development Administration to prepare a report on the vehicle test program and to describe the preliminary design of a smaller (80-100 bhp) 4-98 Stirling engine suitable for a compact car (Reference 4-6). A follow-on contract was signed in 1977 between Ford and the Energy Research and Development Administration (now the Department of Energy, DOE) to develop an improved Stirling engine for automotive application. This cost-shared development program between Ford and DOE will require 8 years for completion and will require a total funding level of \$160 million. Management of this contract has been delegated by DOE to the National Aeronautics and Space Administration, Lewis Research Center, Stirling Engine Project office. This program is outlined in Reference 4-7.

In 1974, a United Stirling V4 X 31 Stirling engine was installed in a Ford Pinto (2500 lb inertia weight) to evaluate vehicle driveability, engine control systems, and cooling systems. More recently, an improved version of this engine (V4 X 35) has been installed in a Ford Taunus station wagon for engine power control evaluation. Efforts at United Stirling of Sweden (USS) are directed toward production of Stirling engines for some specialized markets and the development of technology for volume markets (including truck and automotive). The engines currently under development include the P40 (40 kW), P75 (75 kW) and the P150 (150 kW). In 1977, an initial version of the P75 was installed in a distribution truck for evaluation.

A second Stirling engine effort is being conducted for DOE by a team composed of Mechanical Technology Incorporated, United Stirling of Sweden, and American Motors. This program will help establish the

developmental base of component, subsystem, and system designs; fabrication technology; test experience; and assessment of the cost and marketability of Stirling engines for automotive uses.

4.2 ENGINE CONCEPTS AND CONFIGURATIONS

4.2.1 Basic Thermal Concept

Like all heat engines, the Stirling engine produces power by compressing a working gas at relatively low temperature, adding heat, and then expanding the gas to produce work. As shown in Figure 4-1, a Stirling engine has five separate spaces which are accessible to the enclosed working gas: expansion space, heater, regenerator, cooler, and compression space. The ideal Stirling cycle, shown in the state diagrams in Figure 4-2, is composed of the following four discrete processes:

- 1 - 2 An isothermal compression of the working gas with heat being transferred from the gas in the cooler.
- 2 - 3 A constant-volume heating of the working gas as it passes through the regenerator.
- 3 - 4 An isothermal expansion of the working gas with heat being transferred to the gas in the heater. Work is produced by the engine in this process.
- 4 - 1 A constant-volume cooling of the working gas as it passes through the regenerator.

This ideal Stirling cycle has the same thermal efficiency as the Carnot cycle which operates between the same two temperature limits. There are many factors which cause actual Stirling engines to deviate considerably from this ideal thermodynamic cycle. For the ideal cycle,

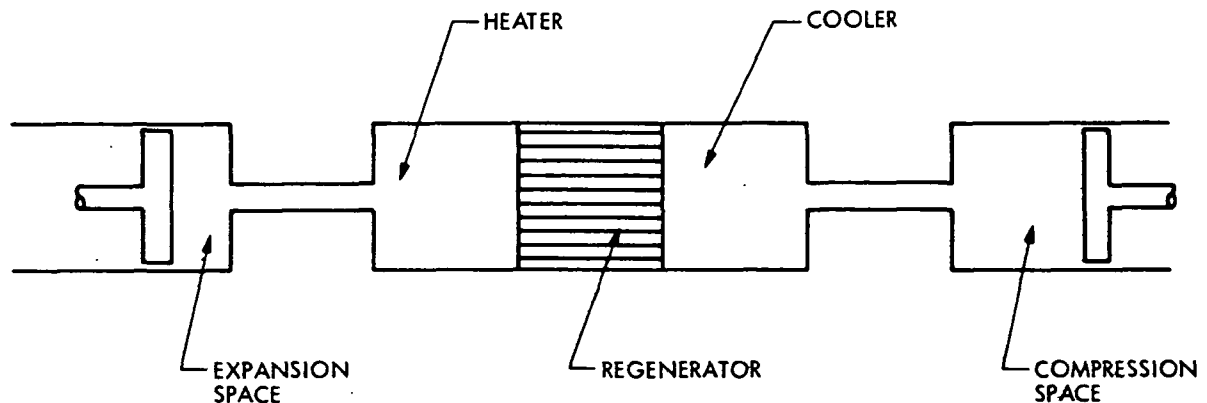


Figure 4-1. Schematic of Stirling Engine

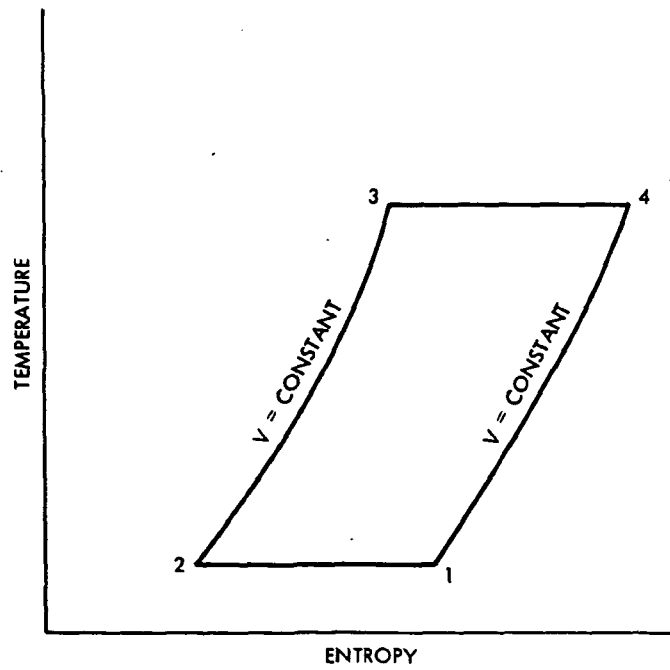
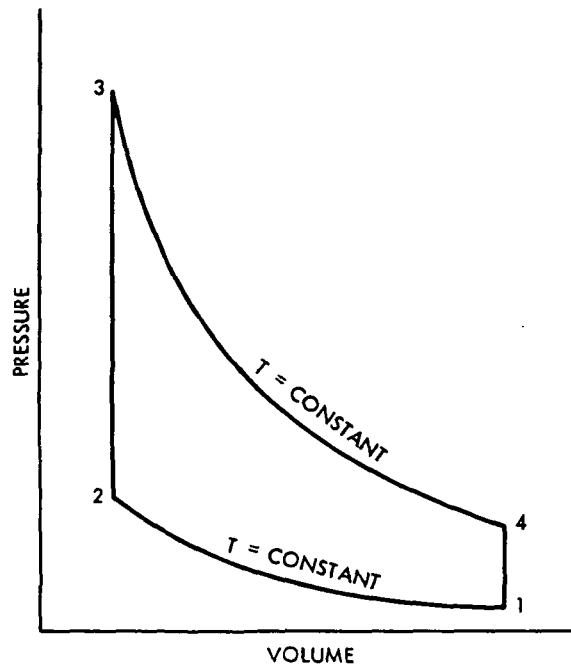


Figure 4-2. State Diagrams for Ideal Stirling Cycle

it is assumed that compression and expansion processes are isothermal, which implies infinite rates of heat transfer between the working gas and the confining cylinder walls. All of the working gas is assumed to be in the compression space during compression and in the expansion space during expansion, which implies zero dead volume (volume of the total working space not in the expansion or compression space). Pistons are assumed to move in a discontinuous manner to achieve the desired movement of the working gas. In an actual engine, the working gas moves continuously from one space to the next and thus undergoes a somewhat different thermodynamic cycle. Perfect regeneration is assumed which implies infinite heat transfer rates between the working gas and the regenerator matrix and infinite heat capacity of the matrix. All friction losses are neglected. When hardware limitations and implementation constraints are considered, the thermodynamic cycle of the working gas in an actual Stirling engine is quite different from the ideal cycle, as is shown in the typical state diagram in Figure 4-3.

The thermodynamic treatment of the Stirling cycle given here has been very brief. More detailed treatments can be found in References 4-1, 4-8, and 4-9. The most comprehensive analysis capability for Stirling engines resides with N. V. Philips as a result of their involvement with Stirling engine development for many years and their access to the most extensive collection of data available from actual engine tests. Unfortunately, most of the analysis capability has not been published in the open literature and is available only to licensees of N. V. Philips. In an attempt to broaden the understanding of Stirling engine analysis and design, DOE is sponsoring several research and technology programs to develop independent (free from licensing agreements) performance and design models for the Stirling cycle (References 4-9, 4-10 and 4-11).

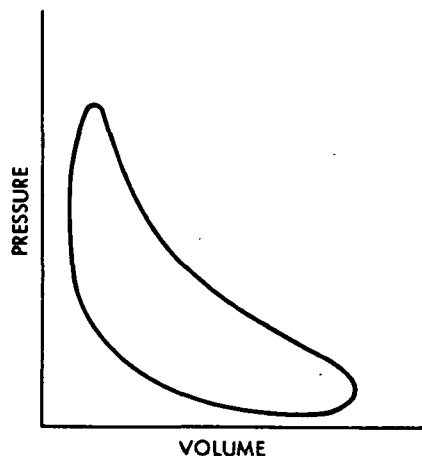


Figure 4-3. Typical State Diagram for Actual Stirling Engine

4.2.2 Engine Configurations

Out of the many possible ways of implementing the Stirling cycle (Reference 4-1), two basic piston mechanisms have received the most development effort. These are the piston-displacer concept and the two-piston arrangement. The piston-displacer concept uses two pistons which can either be located in the same cylinder or in separate cylinders. A sketch of the concept in a single cylinder is given in Figure 4-4. The power piston extracts the useful work of the cycle during expansion and provides the work of compression. The displacer piston causes the working gas to flow alternately to the expansion space (hot space) and to the compression space (cold space) through the heater, regenerator, and cooler. The displacer and power pistons must be connected with linkages in such a way as to phase the movements needed in the Stirling cycle.

Power removal from piston-displacer engines has been accomplished using crankshafts and rhombic drives. In the early years after 1938, many of the small Stirling engines built by Philips were single-cylinder, piston-displacer engines which used crankshafts. The crankshaft designs and the state of seal technology at that time required the use of pressurized crankcases in the engines to compensate for piston forces and to limit leakage. Pressurized crankcases imposed limits on acceptable working gas pressures and thus also on the power output per unit engine displacements making these designs unsuitable for large engines. The invention of the rhombic drive in 1953 made it possible to

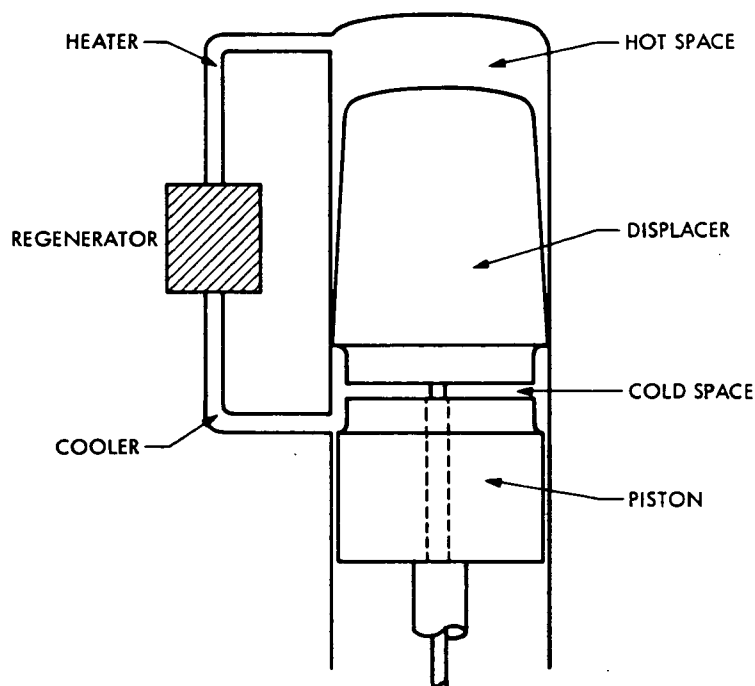


Figure 4-4. Single-Cylinder Piston-Displacer Stirling Engine Concept

design a piston-displacer engine with a nonpressurized crankcase (Reference 4-12). The rhombic drive permitted a single-cylinder engine to be fully balanced. The new drive system also made straight-line movement of the rods possible so that smaller diameter external seals could be used. The introduction of the roll-sock seal (Reference 4-13) in 1960 aided the further development of Stirling engines.

Many piston-displacer engines with rhombic drives have been built and tested by Philips (Reference 4-14) and their licensees. One of the most noteworthy engines of this type is the Philips 1-98 test engine (single cylinder), which was first constructed in 1959 and has undergone several redesigns. These engines could be built in multicylinder versions because of their modular design. Multicylinder versions have been installed and tested in buses at Philips (4-235 engine, Reference 4-15) and United Stirling (4-615 engine, Reference 4-16). The piston-displacer engine with rhombic drive is not currently being considered for automotive application because of its complexity (with a combustion chamber, two pistons, and two rod seals for each cylinder) and its low power density.

The two-piston engine concept can be implemented in either the single-acting (Figure 4-1) or the double-acting (Figure 4-5) arrangement. In the single-acting configuration, one piston acts as the power piston and moves the working gas from the hot space, through the heat exchanger, and to the cold space. The second piston acts as the compressor piston and moves the working gas from the cold space, through

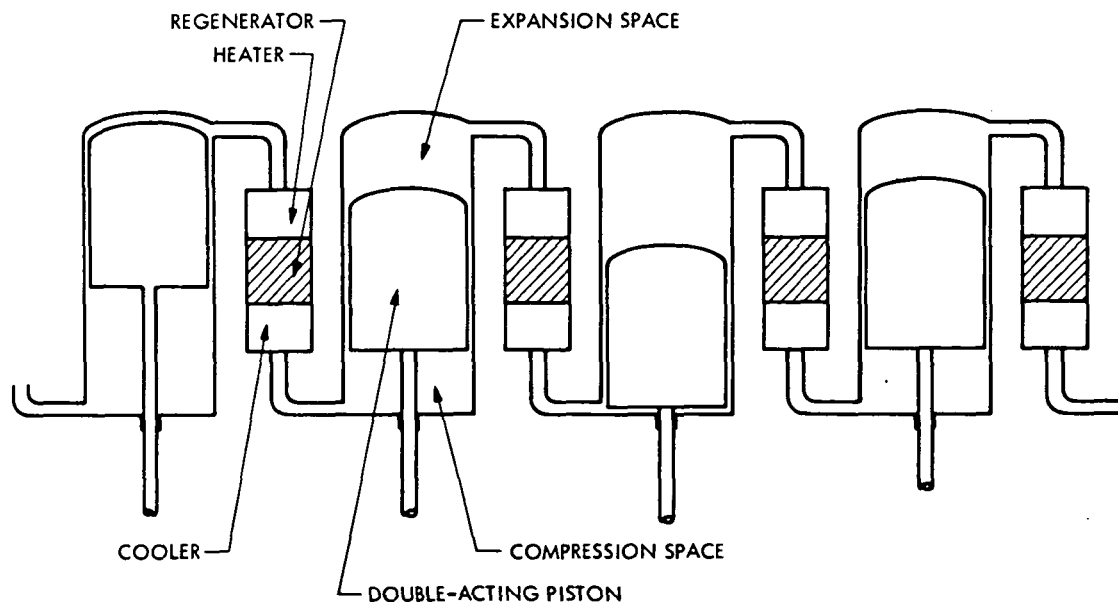


Figure 4-5. Double-Acting Two-Piston Stirling Engine Concept (Rinia Arrangement)

the heat exchangers, and to the hot space. In the double-acting (DA) version, several cylinders are interconnected as shown in Figure 4-5. The upper face of a piston bounds an expansion space which is connected through a heater, regenerator, and cooler to a compression space bounded by the lower face of an adjacent piston. This arrangement is known as the Rinia configuration (Reference 4-17).

The DA engine eliminated the need for a pressurized crankcase and reduced the number of pistons and piston rod seals by a factor of two compared with the piston-displacer engine. However, the DA engine requires multiple cylinders and is more complex from a thermodynamic and fluid dynamic point of view than the piston-displacer engine, which provides a more complete separation of cycle functions. Because of the potential Stirling engine applications being considered at the time, work on the DA engine was stopped when the rhombic drive for the piston-displacer engine was invented.

As interest in applying Stirling engines to mobile applications grew, investigations of DA engines were again started at Philips in 1964. General Motors began an investigation of the Rinia configuration in 1965 for a torpedo propulsion system (Reference 4-3). Improvements in seal technology and advancements in the understanding of the thermodynamics and flow in Stirling machines made the capabilities of the DA engine concept look more attractive. Today, the DA engine configuration is the only serious approach being considered for automotive Stirling engines.

Many DA Stirling engines have been built and evaluated by Philips and its licensees. These have included engines with both swashplate and crankshaft drive mechanisms. Philips began the design of the 4-65 DA engine with swashplate drive in 1968. This engine was the forerunner of the Ford/Philips 4-215 DA engine. The swashplate drive was chosen for the 4-215 engine since an engine of this design could be conveniently packaged in the test vehicle, a Ford Torino. Studies (Reference 4-6) have shown that this configuration may not be the optimum choice for smaller compact vehicles.

Work on DA Stirling engines was started at United Stirling of Sweden (USS) in 1972 with the V4X engine, which was a spin-off from the Philips 4-65 engine with swashplate drive. The V4X engine used essentially the same components as the 4-65 engine except the V4X engine used a crankshaft for power removal. Different versions of the V4X engine have been installed in a Ford Pinto and a Ford Taunus station wagon for testing of certain engine functions. The engines currently under development at USS include the P40, which is a U-design with dual crankshafts, and the P75 and P150, which are V-designs with single crankshaft drive mechanisms. United Stirling has also built the P75 engine in the U-design configuration with dual crankshafts.⁽¹⁾

⁽¹⁾ See Note 1, pg. 4-54.

In the previous section the basic thermal concept of the Stirling cycle and the most important engine configurations were discussed. In this section developments of the major components and subsystems which make up a Stirling engine will be covered. Only double-acting (DA) two-piston Stirling engines which use hydrogen gas as a working fluid and have continuous combustor heat sources will be discussed since all of the engines being considered for the automotive application fall into this category. The major components making up a Stirling engine are the heater head, combustor, regenerator, air pre-heater, and cooler. Other major subsystems required for an automotive Stirling engine are a drive mechanism, power level control subsystem, mixture control subsystem, hydrogen containment subsystem, and accessory drives.

4.3.1 Heater Head

The heater head assembly acts as the heat exchanger for the transfer of heat from the external heat source to the enclosed working gas of the Stirling engine. This assembly is one of the most difficult design and fabrication problems in the engine because of its operating environment and other design requirements. Most of the present heater head designs consist of a series of small-diameter tubes through which the high-pressure hydrogen working gas passes in going from the regenerator to the expansion space. Because the working gas is hydrogen, the tubes must be resistant to hydrogen embrittlement. The tubes must operate at the peak cycle temperature, which must be as high as possible, for best engine thermal efficiency. On their external surfaces, the tubes are exposed to the products of combustion from the combustor and therefore are subject to high-temperature oxidation and corrosion. At this high operating temperature, the tubes must resist loss of hydrogen working gas by diffusion and must also introduce a minimum pressure drop as the working gas flows through them. The heater tubes contribute to the dead volume of the engine and therefore the internal volume of the heater tubes should be a minimum for best engine thermal efficiency. In addition to these difficult requirements, the heater head assembly must be durable, reasonable to manufacture, and not too costly.

Engine controls are set to maintain the heater head temperature constant for all engine operating conditions. This is done primarily for three reasons (Reference 4-16):

- (1) To minimize thermal fatigue of heater tubes due to temperature cycling,
- (2) To maximize engine thermal efficiency by operating at maximum allowable temperature for all engine operating conditions, and
- (3) To avoid a response lag during load changes due to thermal inertia of the engine.

Present Stirling engine heater heads are made from superalloys, and their operating temperatures are limited by metal technology. The Ford/Philips 4-215 engine uses a heater tube temperature of 750°C (Reference 4-15), while the United Stirling P75 maintains a 720°C heater tube temperature (Reference 4-18). Increases in heater tube temperature to 770-820°C (expansion gas temperatures 700-750°C) are considered compatible with the use of improved metallic materials. To achieve the expansion gas temperatures (1000-1100°C) envisioned for advanced Stirling engines will require the use of ceramics, such as silicon carbide and silicon nitride. Ceramic technology is discussed in Section 8.

Although it is desirable to maintain all heater tubes at the same constant temperature, this goal is difficult to achieve in practice. Temperatures in the heater tubes of the Ford/Philips 4-215 engine vary from 620°C to 820°C in the first row and from 660°C to 680°C in the second row.⁽¹⁾ Factors which contribute to these variations include hot spots, poor mixing, and asymmetry in the exit flow from the combustor.

The need to minimize hydrogen diffusion losses through the hot heater tubes has prompted considerable work on hydrogen diffusion barrier coatings. Philips estimates that about 10-15 percent of the hydrogen leakage in the 4-215 engine occurs through the engine hot parts with the remainder being lost through the roll-sock seals.⁽¹⁾ Although Philips has successfully tested barrier coatings on single heater tubes, application of these coatings to complete heater head assemblies has not been satisfactory.

The most efficient heater head designs from a performance standpoint have been rather complex and difficult to build. United Stirling has spent considerable effort in working on simpler techniques for the fabrication of heater head assemblies. Two involute type heater head configurations, which are being evaluated by United Stirling for use in their P series engines, are shown in Figure 4-6. The Philips heater head design for the 4-215 engine is similar to the United Stirling involute design, except individual heater tubes can be replaced on the United Stirling design, while a minimum of three heater tubes must be replaced in the Philips design.

4.3.2 Combustor

The combustor in a Stirling engine must continuously supply hot exhaust gases which are used to heat the enclosed working fluid in the heater head assembly. Since it is desirable to maintain a constant heater head temperature, the flow rate through the combustor must be varied depending on the power required from the engine. To provide equal heating to all heater tubes, the combustor must supply a uniform exhaust gas flow over the heater tubes. This requires that the combustor have good fuel atomization, promote uniform mixing of fuel, air,

⁽¹⁾ See Note 2, pg. 4-54.

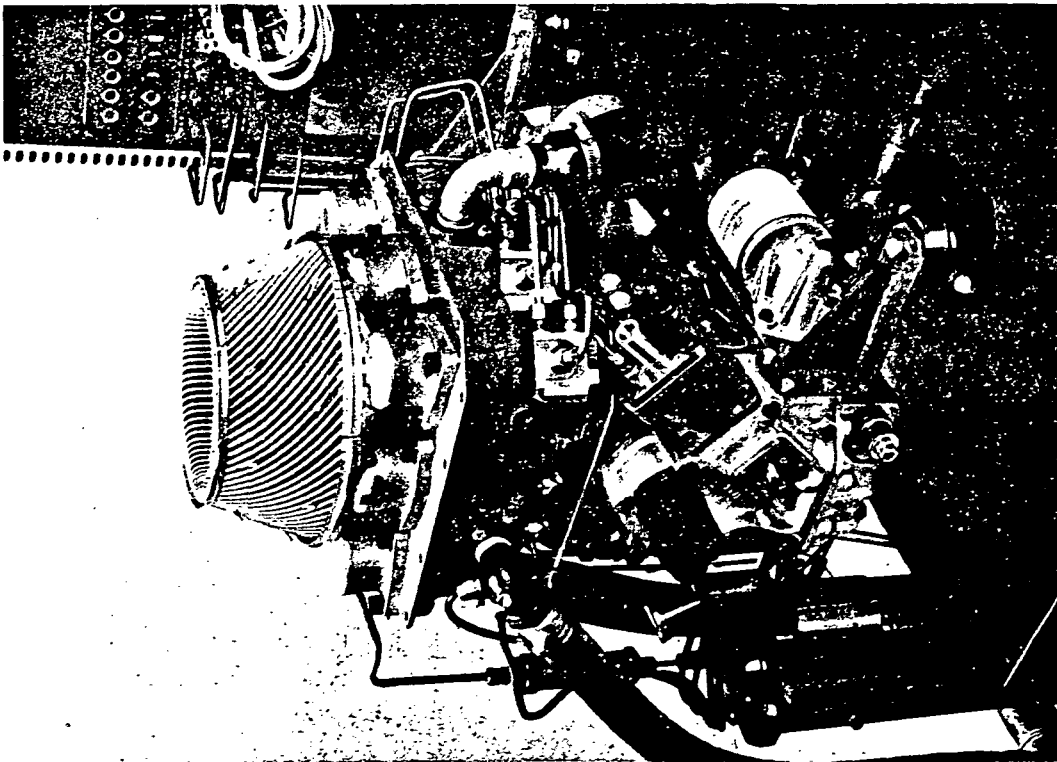
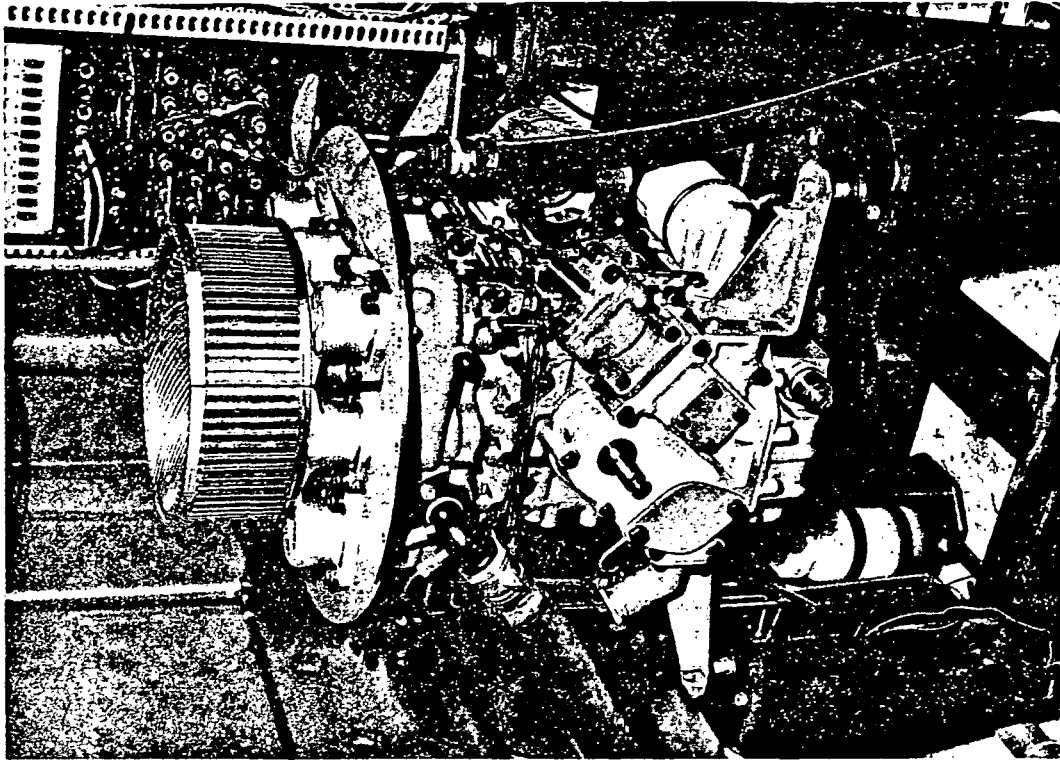


Figure 4-6. Involute Type Heater Head Configurations
(United Stirling)

and EGR to eliminate hot spots, and provide a uniform flow over the heater tubes for all engine operating conditions. The interaction of the exhaust gas with the heater tubes depends on the combustor geometry and on the geometrical relationship between the combustor and the heater tubes.

While supplying energy to the working fluid, the combustor must accomplish the combustion of the fuel in such a way as to minimize the HC, CO, and NO_x emissions in the exhaust stream. This requires control of the equivalence ratio and EGR flow rate for all engine operating conditions. Most combustors employ a two-stage combustion process to achieve the required low emissions. The NO_x reactions are essentially frozen by providing a short residence time at high temperature in the primary zone of the combustor. A longer residence time is provided in the dilution zone at a lower temperature (below the NO_x formation temperature and above the CO conversion temperature) for the conversion of CO.

Combustors for Stirling engines have in general been of the standard fixed-geometry type consisting of a burner can, fuel atomizer, and igniter. The double-acting (DA) engines have normally been arranged so that a single combustor unit is used to supply energy to four cylinders. As previously noted, considerable variations in individual heater tube temperatures have been experienced with these designs.

Several auxiliaries are used to improve the efficiency of the combustor. These include the air preheater, combustion air blower, and EGR system. The air preheater and blower are discussed in another section of the report. Exhaust gas recirculation is used to help reduce the amount of NO_x emissions in the exhaust. The first Stirling engine to use EGR was the Ford/Philips 4-215 engine. Large amounts of EGR have been required in the 4-215 engine (up to 130 percent according to Ford).⁽¹⁾

In advanced Stirling engines, which will operate at much higher temperatures, it may be necessary to use more complex combustor designs to achieve the low NO_x emissions level required.

4.3.3 Regenerator

The regenerator in a Stirling engine is an energy-storage heat exchanger located between the heater and cooler. Its function is to remove heat from the working fluid as it moves from the expansion space to the compression space, store the heat in the regenerator matrix, and return the heat to the working fluid as it moves from the compression space back to the expansion space. In double-acting Stirling engines, one regenerator is required in each flow path connecting a compression space with an expansion space. The effectiveness of the regenerator depends on the matrix design, the mesh size and spacing, and the material's thermal conductivity and heat capacity. Most regenerators

⁽¹⁾ See Note 3, pg. 4-54.

have used closely packed fine-mesh screens or monolithic structures for energy storage. Regenerators need a large heat capacity for energy storage, a high thermal conductivity, and small flow passages to ensure an adequate surface film coefficient for heat transfer. It is desirable that the void volume in the regenerator matrix be small since this space contributes to the engine dead volume. However, smaller flow passages produce larger flow pressure drops, reducing engine efficiency. Since the regenerator is located between the heater and cooler, the regenerator matrix and container should function as a thermal barrier in the flow direction.

These somewhat conflicting requirements make the design of Stirling engine regenerators a difficult task. Although considerable data exist for energy-storage heat exchangers, little data is available for the operating conditions encountered in Stirling engines (i.e., hydrogen working gas, high pressure, high cyclic operating frequency). The design of regenerators for the Stirling engine application is discussed in Reference 4-1.

Little detailed information has been obtained about the performance of regenerators in actual Stirling engines. The difficulty of making meaningful measurements has prevented any systematic evaluation of regenerator performance; however, no apparent problems have surfaced in the development of the 4-215 engine at Ford.

4.3.4 Air Preheater

The air preheater in the Stirling engine is a heat exchanger which removes heat from the exhaust gas and transfers it to air stream going to the combustor. The more effectively this heat transfer takes place, the higher the thermal efficiency of the engine becomes. The preheated air promotes better fuel vaporization in the combustor, leading to a more complete combustion of the fuel-air mixture. Two types of heat exchangers have been used in the past, and both are still being considered for the Stirling engine application. These are the rotary air preheater and the recuperator.

The rotary air preheater is a storage-type heat exchanger which consists of a porous disc which rotates at a low speed. The disc is alternately exposed to hot exhaust gases and then to the incoming air supply to the combustor. This design was developed for use as a gas turbine regenerator. The disc can be made from either metal or ceramic; however, most present interest is centered on the ceramic unit. The first Stirling engine with a rotary ceramic air preheater was the Ford/Philips 4-215 engine. The primary advantages of the rotary air preheater are its high effectiveness, high temperature capability, light weight, compact size, and possibly lower cost. The high effectiveness is possible since flow passages of small hydraulic radius are feasible with an acceptable flow pressure drop. Any buildup of deposits is inhibited with this type of preheater by the periodic flow reversal and the alternate exposure of the deposits to an oxidizing environment.

The primary disadvantages of the rotary air preheater are gas leakage, seal wear, drive complexity, and possibly limited durability. It is necessary to provide a rubbing seal which slides along the face of the rotating porous disc. Although considerable seal development has been done, leakage along the seal continues to be a problem at current operating temperatures (about 1850°F for inlet exhaust temperature) and promises to be an even larger problem on advanced engines with higher temperatures. Much of the leakage may be the result of seal wear which occurs due to abrasion and thermal distortion of the porous disc. At the present time, a durability of over 4000 cycle hours has been demonstrated with a gas turbine ceramic regenerator of a similar design; however, further testing is still required before the design is adequate for automotive use. The durability of the rotating design is impaired by the continuous action of thermal stresses and corrosion which can lead to cracks in the disc. Rim drives are currently used for driving the rotating air preheater.

The recuperator-type air preheater is a fixed boundary heat exchanger for transferring heat from the exhaust gas to the incoming air by conduction through the walls of the recuperator. Current recuperators are made from metal; however, there is interest in using ceramics in advanced recuperators to increase their operating temperature and reduce weight and cost. Stirling engines being developed by United Stirling of Sweden (the V4X, P40, P75, and P150 engines) use metallic recuperators. Although current operating temperatures are less, United Stirling estimates that metallic recuperators could be used for exhaust gas temperature of 900°C with metal surface temperatures in the 700-800°C range. The primary advantages of the recuperator are that it has no leakage, no seal wear, and no drive mechanism. The primary disadvantages are a lower effectiveness, larger weight, larger size, lower operating temperature, and the possibility of deposit buildup. To keep the pressure drop, weight, and size to acceptable levels, the recuperator must be designed with fairly large flow passages, and this leads to lower effectiveness values. More advanced ceramic recuperators may help reduce weight and cost. The possibility of a buildup of deposits is more serious in the recuperator since it is not a self-cleaning design. Deposits would tend to reduce the effectiveness of the heat transfer surfaces. Preheaters with high effectiveness values can reduce the temperature of the exhaust gases below the dew point of sulfur compounds (200-250°C), leading to corrosion of metallic recuperators at some engine operating conditions. Although this would reduce the sulfur compounds present in the exhaust gas, the deposits would reduce the effectiveness of the heat exchanger.

Although the trade-offs between recuperators and rotary air preheaters are complex, indications are that the rotary air preheaters will be a better design for advanced high-temperature Stirling engines. A discussion of the ceramic materials which are appropriate for rotary air preheater (gas turbine regenerator) application is given in Section 8 of this report. In a materials technology assessment for Stirling engines which is being conducted by NASA/Lewis Research Center and the Army Materials and Mechanics Research Center (Reference 4-19), the air preheater is considered one of the most critical components for the advanced Stirling engine. The temperature (up to 2700°F) and time

(3500 hours) requirements for this engine far exceed the available test data on ceramic materials such as silicon carbide and silicon nitride. In the ceramic regenerator work for gas turbines being done by Ford Motor Company for DOE, several ceramic regenerators have been tested at 1000°C (1832°F) for over 4000 hours without failure (Reference 4-20). Although progress is being made, there is clearly a need for additional work on advanced ceramic materials to support the advanced Stirling engine development.

4.3.5 Cooler and Cooling System

The cooler is a heat exchanger, located between the regenerator and the compression space. When the working fluid is being compressed, the cooler removes heat from the working fluid so that the process approaches an isothermal compression. The cooler consists of a number of small, high pressure tubes which are surrounded by jacket cooling water. The heat that is removed from the working fluid by the cooling water is rejected to the atmosphere in the radiator. The operation in the cooler is similar to that in the heater tubes except the temperature level in the cooler is much lower and the cooler uses a liquid heat transfer medium on the coolant side. Small diameter tubes are needed to keep the Reynolds number high enough for effective heat transfer from the working fluid. Total volume in the cooler should be small to minimize the dead volume of the engine; however, there must be adequate surface area to accomplish the necessary heat transfer. The flow area in the cooler tubes must be large enough to produce a small pressure drop. In double-acting (DA) engines, one cooler is required in each flow path connecting a compression space with an expansion space.

The thermal efficiency of a Stirling engine is very sensitive to the compression gas temperature, which is maintained by the cooler. A low cooling water temperature is necessary under all operating conditions for good efficiency. Also, Stirling engines reject twice as much heat to the cooling system as do equivalent Otto or diesel engines. In Otto and diesel engines, more energy is lost in the exhaust. The fact that Stirling engines must reject more heat at a lower temperature than conventional engines places increased importance on having an adequate cooling system.

Because of the higher heat load, the cooling system must include a radiator with more heat transfer area and a larger fan. Truck radiators with twice the capacity needed by a conventional engine of equivalent power have been used successfully. New design approaches include a folded front radiator with a finer fin structure and a thinner core than conventional radiators. The larger cooling fan, when it is in use, imposes a larger parasitic loss for Stirling engines compared with conventional engines.

4.3.6 Power Control System

The function of the power control system in the automotive application is to adjust the engine power to match the power needed for

a particular vehicle maneuver (i.e., acceleration, deceleration, climbing a grade, etc.). Since much automobile operation is in the transient mode, the power control system is a very important factor in the development of automotive Stirling engines. Much of the early Stirling engine work at Philips was directed toward applications with little or no transient requirement; therefore, work on the power control system has been a recent development.

The primary types of power control systems for Stirling engines are (1) mean-pressure-level (MPL) control, (2) dead volume control, (3) stroke control, (4) intermittent short-circuit control, and (5) phase control. The phase control method will not be discussed here since it is applicable only to single-acting engines, and these are not being considered for the automotive application. All of the other methods can be used with double-acting Stirling engine designs.

In the mean-pressure-level (MPL) control method, the power output of the Stirling engine is controlled by varying the mean cycle pressure of the working gas. A schematic of the MPL control used by United Stirling is shown in Figure 4-7. The system consists of a

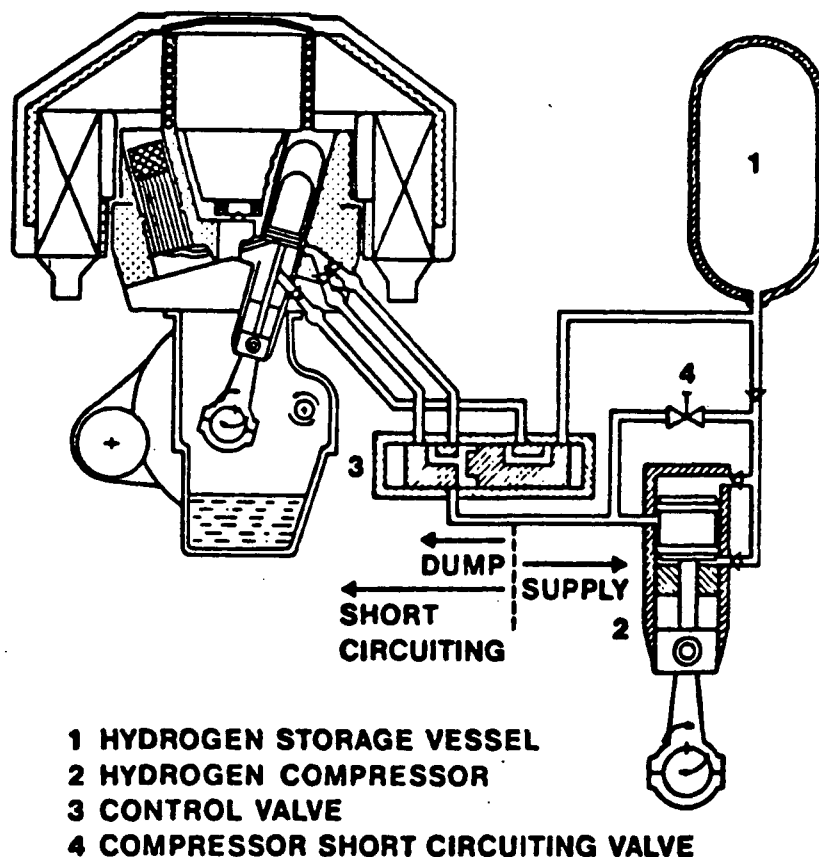


Figure 4-7. Mean-Pressure-Level Power Control System (United Stirling)

servo-actuated control valve, gas distributor, storage cylinder, and compressor. When the engine load is increased and more horsepower is required, the control valve allows additional working gas to flow from the high-pressure storage cylinder into the engine working volume. The distribution mechanism sequences the flow such that the additional working gas enters the engine at the correct time when the cycle pressure is near its maximum value. For reduced engine power, the control valve is positioned so that the compressor will pump some of the working gas back into the storage cylinder. The position of the control valve depends upon the accelerator position and a feedback system which senses the mean cycle pressure. A rapid decrease in power is obtained by short-circuiting the hydrogen gas between cylinders.

Stirling engine systems which have used the MPL control method include the Ford/Philips 4-215 engine and the United Stirling V4X35, P40, and P150 engines. Power control problems which have been encountered in the Ford 4-215 engine program are RF susceptibility, hydrogen leaks, control instabilities, and system contamination.⁽¹⁾ Some of these problems have been corrected; however, the power control instability problem is yet to be satisfactorily resolved (Reference 4-7). One factor which could be responsible for the poorer-than-expected in-vehicle fuel economy measured with the 4-215 engine is the loss associated with decreasing engine power level. Short-circuiting the hydrogen gas between cylinders to quickly drop power results in a fuel economy penalty. Also, the compressor which pumps working gas back to the storage cylinder works more when there is leakage in the power control system. This increased compressor work also hurts fuel economy. The United Stirling V4X35 engine has been installed in a Ford Taunus station wagon for evaluation of its MPL control system. Driving tests have demonstrated that the vehicle has good driveability and adequate engine response.⁽²⁾ Tests of a free-running (disengaged from the dynamometer) engine, equipped with all auxiliaries, has demonstrated a response time of 1 to 1.5 seconds from idle to full power (Reference 4-18).

In the dead volume control method, engine power is controlled by adding or removing volumes available to the working gas and thus modulating the pressure amplitudes in the engine. The mean pressure in the engine is usually held constant while volumes are added or removed in discrete increments through the use of valves. The use of discrete volume elements produces step changes in engine power output; however, the engine power can be made to change more continuously by using more volume elements of smaller volumes. The use of more volume elements and valves makes the dead volume control approach more complex and costly and therefore less attractive.

In 1974, a United Stirling V4X31 engine with dead volume power control was installed in a Ford Pinto to evaluate the engine control system. Although little information is available about these

(1) See Note 3, pg. 4-54.

(2) See Note 1, pg. 4-54.

vehicle tests, United Stirling has selected the MPL control method for its future engines. The reasons given are its good response to load variations, good part-load efficiency, and simple components (Reference 4-16).

In the stroke control method, engine power is controlled by varying the piston stroke and thus modulating the pressure amplitude in the engine. One way of implementing this approach is by means of a variable-angle swashplate (VASP) drive. The VASP drive could be coupled directly to the accelerator pedal. Analyses of the VASP drive have been made, but no data from actual hardware tests has been reported. Philips has plans to build a small Stirling engine with a VASP drive for evaluation purposes.⁽¹⁾

In the intermittent short-circuit control method, engine power is varied by intermittently short-circuiting the different cylinders in a double-acting Stirling engine. This method is simple and gives good load variation response. This power control technique is being pursued by MAN/MWM; however, no information is available on the results of their evaluation.

4.3.7 Mixture Control System

The mixture control system varies the air and fuel flow to the combustor in such a way as to keep both the heater head temperature and equivalence ratio constant over the entire operating range of the engine. This is accomplished by sensing heater head temperature with thermocouples and using the signal to control air and fuel flow. The mixture control depends indirectly on power control, which determines the amount of heat needed by the working fluid. If the engine uses mean-pressure-level power control, more power is produced by adding working fluid to the engine. As working fluid is added, heater head temperature drops. This drop in temperature is sensed by a thermocouple which signals an air valve to supply more air through the blower and into the combustion chamber. An air flow sensor measures the amount of air flow and signals for additional fuel to keep the equivalence ratio constant.

The reasons for controlling the engine to keep heater head temperature constant have been discussed in Section 4.3.1. The control of equivalence ratio is necessary to produce good exhaust emissions. The control system can be designed to tailor the equivalence ratio over the load range.

The first generation mixture control system for the Ford/Philips 4-215 engine is shown in Figure 4-8. It is basically a mechanical system in the early stages of development. Early problems with this unit were related to instabilities in the control circuit. Also,

⁽¹⁾ See Note 2, pg. 4-54.

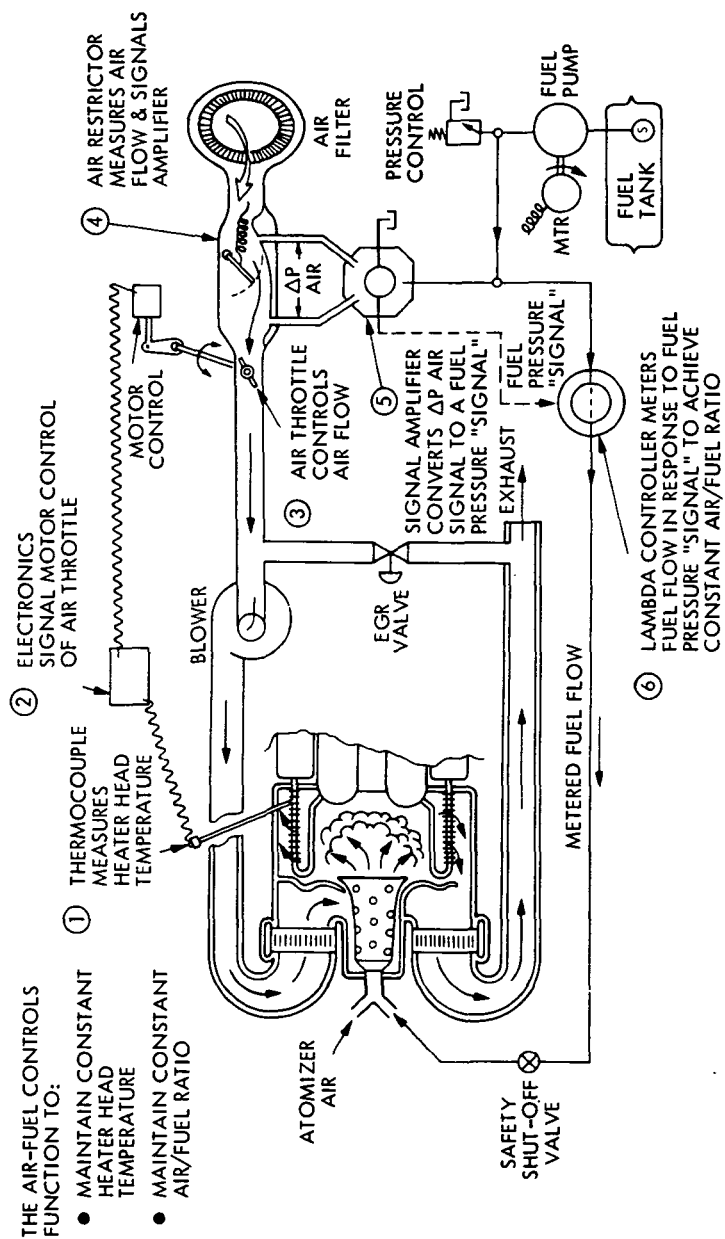


Figure 4-8. Mixture Control System (Ford/Philips 4-215 Engine)

the current air flow sensing device requires a large pressure drop, which increases air blower requirements. The larger blower contributes to larger parasitic losses and lower engine system efficiency. In their most recent engines, United Stirling uses a modified Bosch K-Jetronics fuel injection system (Reference 4-21) to provide mixture control.

4.3.8 Hydrogen Containment Systems

The Stirling engine requires a complex system of seals and surface barriers to contain the high pressure working gas (hydrogen) and to prevent the lubricating oil from leaking into the gaseous working spaces. The critical seal in the Stirling engine is the piston rod seal which is located between the piston and cross-head member. With an unpressurized crankcase, the seal must maintain positive isolation between the high pressure working fluid and the ambient pressure in the crankcase. Two different types of seals are under development, the roll-sock seal and the sliding seal.

The roll-sock seal system (Figure 4-9) is an oil-supported diaphragm type which has a separate, regulated pressure system to minimize the pressure differential across the diaphragm. The roll-sock seal is rigidly attached to both the piston rod and the adjacent wall, providing an hermetic seal. The sealing system includes the hydraulic fluid and its regulating valve, which controls the pressure differential across the roll-sock diaphragm to a few atmospheres. The hydraulic system thus provides support for the roll-sock. The oil is provided by a metal pumping ring which also functions as a seal. The roll-sock does

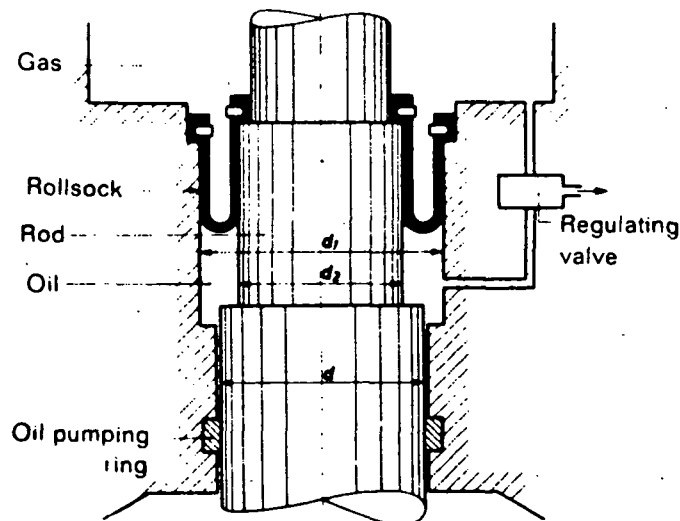


Figure 4-9. Roll-Sock Piston Rod Seal

not stretch during its roll maneuver since the steps in the rod and housing are designed to maintain an equal volume of oil throughout the piston stroke.

Some of the problems associated with the roll-sock seal are as follows: it is easily damaged (especially during installation), degrades with temperature cycling (it is made of elastomeric material), is incompatible with many hydraulic oils and is somewhat permeable to the hydrogen working fluid. Also, the integrity of the seal depends on the hydraulic control system to maintain the small differential pressure across the diaphragm. Failure of this supporting system represents a catastrophic engine failure since the engine has to be completely disassembled to clean the oil from the working gas space. The advantages of the roll-sock seal include its compact design, moderate cost, ability to function satisfactorily at high engine speeds, and its reasonable mean lifetime.

Currently, the roll-sock seal is used in the Ford/Philips 4-215 engine. During the development effort at Ford, failures of the seal system have resulted from problems with the pressure regulation system, poor oil pumping ring performance, improper installation of the roll-sock, and contamination.⁽¹⁾ Consideration is being given to the use of a backup sliding seal to prevent total engine failure when a roll-sock seal problem occurs. The roll-sock seal represents one of the most critical leakage paths in the engine. Philips estimates that in the 4-215 engine 10-15 percent of the hydrogen loss is through the hot parts and static seals and 85-90 percent of the hydrogen loss is through the roll-sock seal.⁽²⁾

The sliding piston rod seal (Figure 4-10) is a compound seal unit which consists of two seal elements, a scraper ring, and oil-gas separator unit. The top seal element converts the oscillating working pressure in the engine cycle to a constant pressure at a level near the minimum cycle pressure. The minimum cycle pressure is maintained by a check valve which brings leaking hydrogen gas from the separator back into the cycle. Lubricating oil that is deposited on the piston rod during its reciprocating motion is removed by the scraper ring, sent to the oil-gas separator, and then returned to the engine oil system. The main seal maintains the pressure difference between minimum cycle pressure and atmospheric pressure in the crankcase. The sliding seal concept described above was developed by United Stirling.

Sliding piston rod seals have been used in most of the engines recently built by United Stirling (e.g., V4X engines, and P40, P75 and P150 engines). United Stirling has tested a sliding seal for several thousand hours, both in component test rigs and in actual engine service. Since the sliding seal has some built-in hydrogen leakage, a periodic recharging of the hydrogen working fluid is required. Sliding seals have demonstrated leakage values of 0.2 normal liters/hour, which

(1) See Note 3, pg. 4-54.

(2) See Note 2, pg. 4-54.

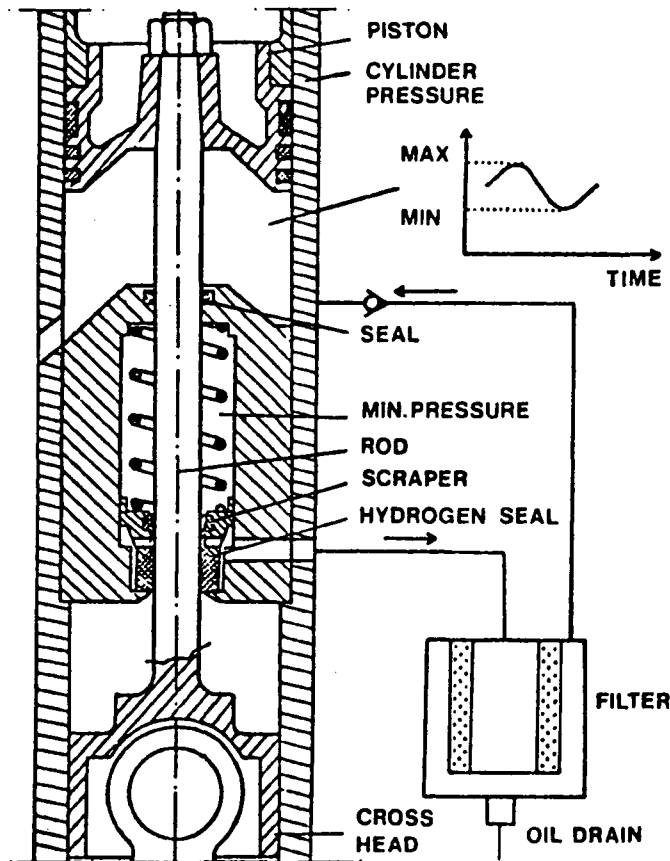


Figure 4-10. Sliding Piston Rod Seal

should permit a 3-liter storage bottle to last about 3 months.⁽¹⁾ Philips and Ford are also working on a sliding piston rod seal of a similar design. Philips has experienced several failures of the scraper ring due to frictional heating of the ring.⁽²⁾ They are concentrating their efforts on finding a better material for the scraper ring. Although much development work remains to be done, the sliding piston rod seal looks attractive as a possible alternative to the roll-sock seal. The sliding seal has a more acceptable failure mode in that the engine fails gradually rather than suddenly.

The hydrogen containment problem is not limited to elastomeric seals alone. Some hydrogen is lost through the metal walls of the engine since hydrogen is able to diffuse through a metallic lattice, especially at elevated temperatures. Proprietary surface coatings developed by Philips are reported to reduce the working fluid loss through the metal parts of an engine by factors of 25 to 50.

(1) See Note 1, pg. 4-54.

(2) See Note 2, pg. 4-54.

4.3.9 Safety

The concern over the potential dangers of fire or explosion with loss of integrity of the system containing the hydrogen working fluid has led to a continuing series of safety tests conducted by the Stanford Research Institute (Reference 4-22). Hydrogen has a high diffusion rate in air with an upper limit of flammability of 75 percent, which is above the upper detonability limit of 59 percent. As a consequence, the observed slow hydrogen leaks do not produce ignitable mixtures. During a rapid escape of gas, such as might occur in a collision of the vehicle, the hydrogen loss could result in a rather minor fire.

4.4 PRESENT STATUS OF DEVELOPMENT OF STIRLING ENGINES

4.4.1 Introduction

In the previous sections, the status of development of the various components of the Stirling engine are discussed in some detail. In this section, the characteristics of complete engines and vehicles powered by state-of-the-art and advanced Stirling engines are considered. To the extent possible, the results given in this section are based on actual test data from prototype engines, but in many instances it was necessary to base the projections on design calculations, extrapolations, and generalizations of the limited test data available, and on computer simulations.

4.4.2 Engine Design and Development Programs

As noted previously, there are currently two groups actively involved in the development of Stirling engines for automotive (passenger car and truck) applications: the Ford Motor Co. in the United States and United Stirling in Sweden. Both of these companies are licensees of N.V. Philips of The Netherlands. In addition, Ford has a licensing agreement with United Stirling. The approaches taken by Ford and United Stirling are quite different. In the case of Ford, the main emphasis has been on the adaptation of existing Philips engine designs to particular models of Ford passenger cars (e.g., Torino and Pinto). Ford has performed some component development studies, but in nearly all cases it was to correct problems uncovered during the testing of the engine received from Philips. The baselines for both the engine and vehicle performance in the Ford program were the characteristics of current Ford products. At United Stirling, many of the problems associated with Stirling engine development have been reexamined, and more of their effort has been devoted to component R&D. United Stirling has not tied their program to developing an engine for a particular vehicle and have used vehicles as test beds to confirm the validity of designs developed on the engine dynamometer.

As a consequence of their different approaches, the Ford and United Stirling engines are quite different. This can be seen from Table 4-1, where the characteristics of the engines being developed by

Table 4-1. Characteristics of Stirling Engines
for Automotive Applications

Developer/Designation:	Philips/Ford		Ford/4-98	United Stirling/		United Stirling/ P40
	4-215			P75		
Status	Test prototype	Design	Test prototype	Test prototype		Test prototype
Type	Double-acting	Double-acting	Double-acting	Double-acting		Double-acting
Piston Configuration/ Number of Cylinders	Horizontal- axial/4	Horizontal- axial/4	Horizontal- axial/4	V/4		U/4
Drive System	Swashplate	Swashplate	Swashplate	Crankshaft		Dual crankshaft
Seals	Roll-sock	Roll-sock	Roll-sock	Sliding		Sliding
Working Fluid	H ₂	H ₂	H ₂	H ₂		H ₂
Max. Pressure (atm)	200	200	200	150		150
Max. HP/RPM	170/4250	84/5000	90/2500	90/2500		55/4800
Max. Torque/RPM (ft-lb)	300/1400	123/1750	230/1000	230/1000		110/800
Working Temperature Limits						
Heater Tubes (°F)	1300	1382	1300	1300		1330
Cooler (°F)	175	176	160	160		160
Efficiency at Max. HP, %	24 (test)	30.5 (calc.)	25 (test)	25 (test)		25 (calc.)
Max. Efficiency, %	32 (test)	34 (calc.)	33 (test)	33 (test)		36 (calc.)

Ford and United Stirling are summarized. The four engines cited are those on which the bulk of the work has been done and on which considerable information is available. The major differences in the designs are cylinder configuration (horizontal - axial vs. V), drive mechanism (swashplate vs. crankshaft), and hydrogen seals (roll-sock vs. sliding). United Stirling is currently testing engines which use a U cylinder configuration with dual crankshafts. This configuration leads to a more compact heater head and offers greater flexibility in the positioning of the output shaft from the engine. Sketches of the various Ford and United Stirling engine configurations are shown in Figure 4-11. It is not yet clear which of the different Stirling engine design approaches will prove to be the most advantageous for automotive applications.

4.4.3 Engine Characteristics

4.4.3.1 Size and Weight. In the passenger car application, the size and weight of the engine as well as its durability are very important. The weight characteristics of Stirling engines are given in Figure 4-12, which shows engine weight as a function of horsepower. Using current weight technology, the data in the plot represents some current developmental Stirling engines. In general, the Ford/Philips engines, being tailored for passenger car applications, are much lighter for the same horsepower than are the United Stirling engines, which have been designed more for truck applications, with greater emphasis placed on durability.

The Stirling engines currently being developed and tested are engineering prototypes and for that reason tend to be heavier than optimized designs for production. The weights of the various subsystems of the Ford 4-98 engine are given in Table 4-2. Weight reductions in each of the subsystems is expected as development of these engines continues. The projected weight characteristic for fully developed Stirling engines is shown in Figure 4-12 along with the data for current Stirling engines. This characteristic has been used in making the fuel economy projections for vehicles with Stirling engines. Stirling engine weights are compared with those for other engine types in Section 9.

The size of an engine is very important from a packaging standpoint. The dimensions (L x W x H) of several Stirling engines are given in Table 4-3. These characteristics are compared with those of other engines in Section 9.

4.4.3.2 Torque and Power Curves. The shape of the torque-speed characteristic is important in determining the acceleration capability of a vehicle. Torque curves for the Stirling engines under development are given in Figure 4-13. The engine torque and speed are normalized with respect to the torque and speed at maximum power. The torque curves are of special interest because they illustrate clearly the high torque of the Stirling engine at low engine speeds. This is typical of any closed cycle engine in which the torque is modulated by changing the

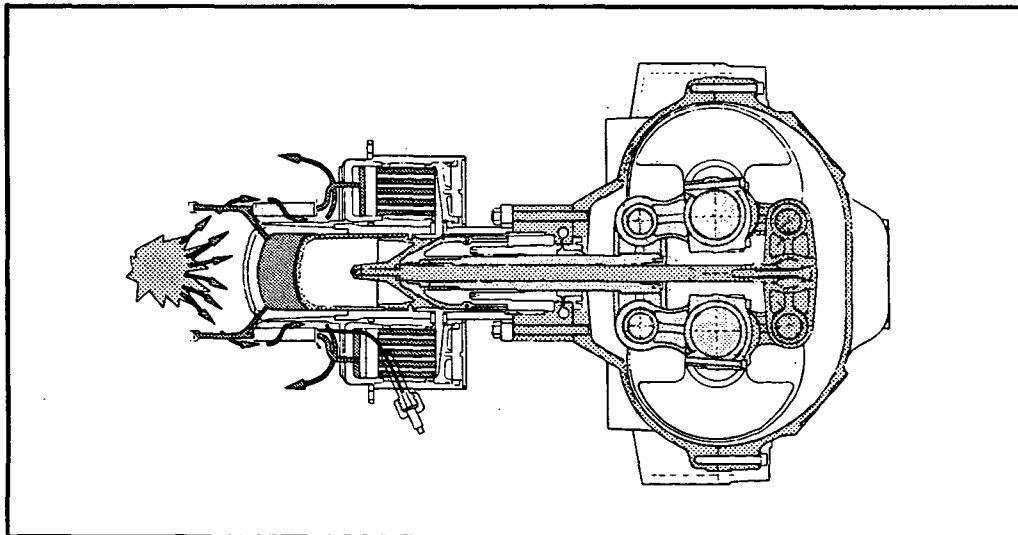
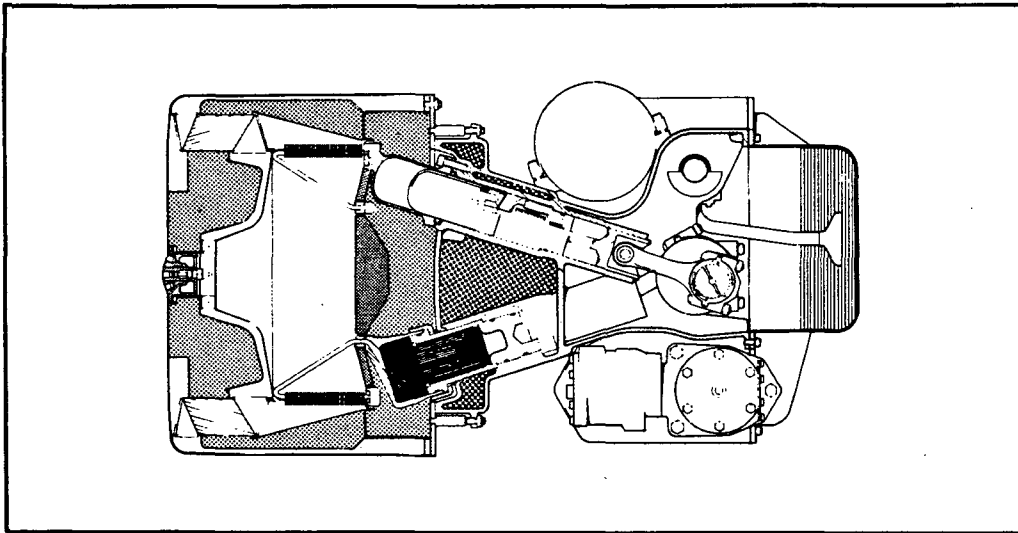
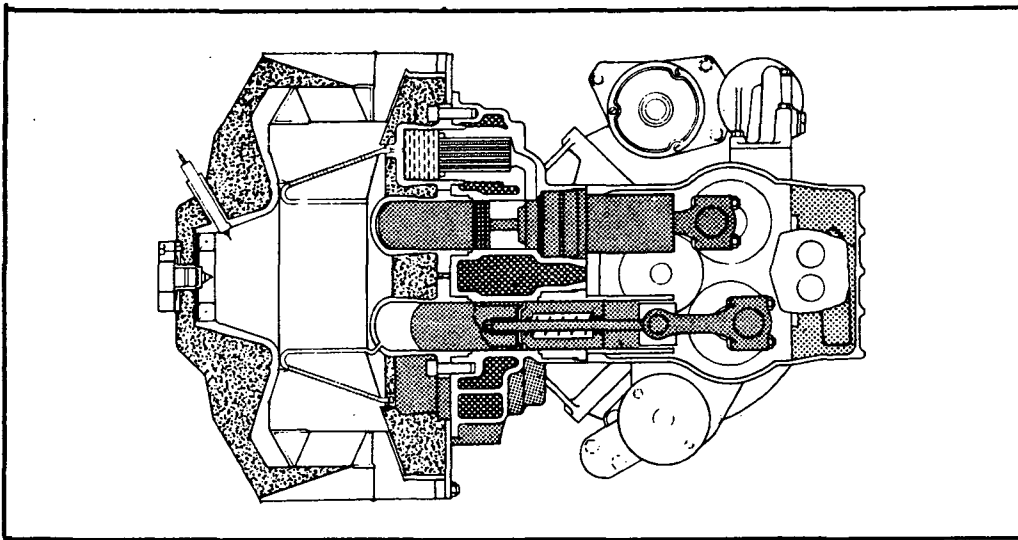
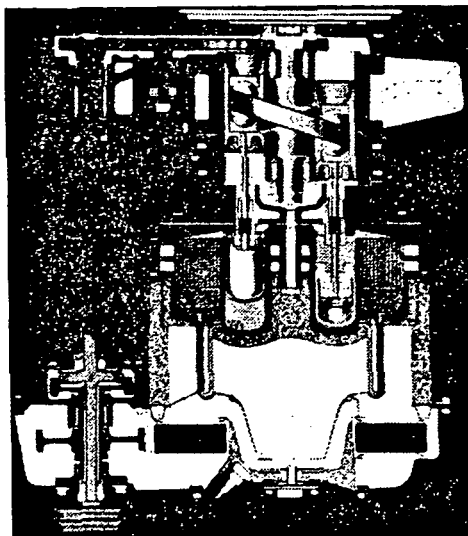
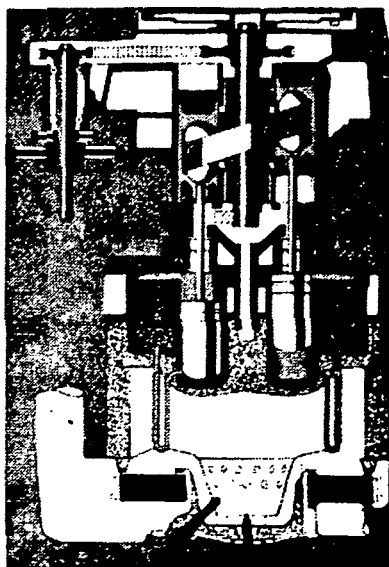


Figure 4-11. Sketches of the Stirling Engine Configurations - United Stirling

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4-98 ENGINE



4-215 ENGINE

Figure 4-11. Sketches of the Stirling Engine Configurations
(Continuation 1) - Ford Stirling

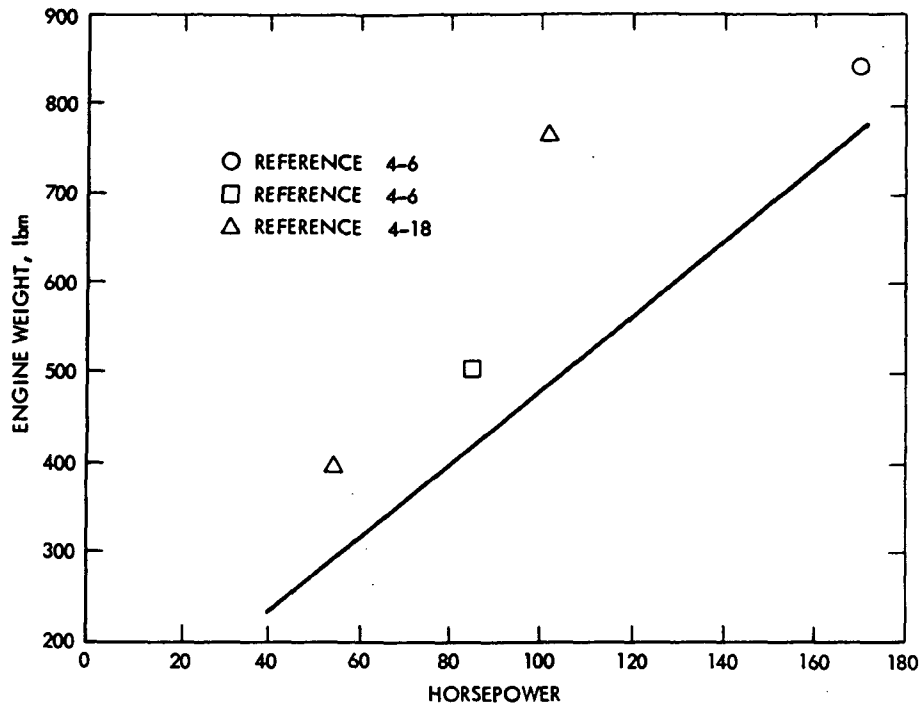


Figure 4-12. Engine Weights vs. Horsepower for Stirling Engines

pressure of the working fluid. The solid line in the figure represents the torque-speed characteristic which has been used in making the fuel economy projections. The torque-speed characteristics of Stirling engines are compared with those of other engine types in Section 9. The corresponding power curves for the Stirling engines under development are given in Figure 4-14.

4.4.3.3 Accessories. A number of accessories are needed to operate the Stirling engine. These include the combustor blower, air preheater motor, radiator fan, and water pump. The power absorbed by the accessories, especially the combustor blower and radiator fan, can be quite high and thus careful attention must be given to meeting the accessory demands efficiently. The combustor blower must handle all of the gases (air plus recirculated exhaust gas) required by the combustor. The power required by the blower can be 5-6 percent of the maximum engine power at high RPM. The cooling system for the Stirling engine must be more extensive than for a conventional engine of the same rated power, because the Stirling engine utilizes a regenerative cycle with most of the waste heat being rejected into the cooling water. This is shown in the table below, in which the energy splits between shaft work, exhaust gases, and coolant are given for typical Stirling, gasoline, and diesel engines. Hence it can be expected that the power needed by the radiator

Energy Split Fractions

Engine Type	Shaft Work	Fraction (%)	
		Exhaust Gas	Coolant
Stirling	36	14	50
Gasoline	28	42	30
Diesel	36	44	20

fan and water pump would be greater in the Stirling engine. Accessory loads for Stirling and gasoline engines are given in Table 4-4 as a function of engine RPM. It is seen that the total accessory loads for the Stirling engine are considerably greater than those of the conventional gasoline engine. The accessory loads for both engines can be reduced significantly by using constant speed drives or at least modulating the drive speed as accessory demand permits.

Table 4-2. Weight Distribution in the
Ford 4-98 Stirling Engine

Engine Section	Weight
Preheater (burner, recuperator, water pump)	75
Heater head (base, tubes, regenerators, cylinders)	115
High pressure crankcase (reciprocating parts, power control)	70
Engine drive (swash-plate, flywheel, oil pump, second block, blower)	100
Accessory drives	40
Radiator (including water)	70
TOTAL	470 lb

Table 4-3. Size Characteristics for Stirling Engines

Engine	hp	Dimensions, in.		
		L	W	H
Ford/Philips 4-215	170	37	26	27
Ford/Philips 4-98	84	26	21	23
United Stirling P40	55	24	--	31
United Stirling P75	100	28	--	36

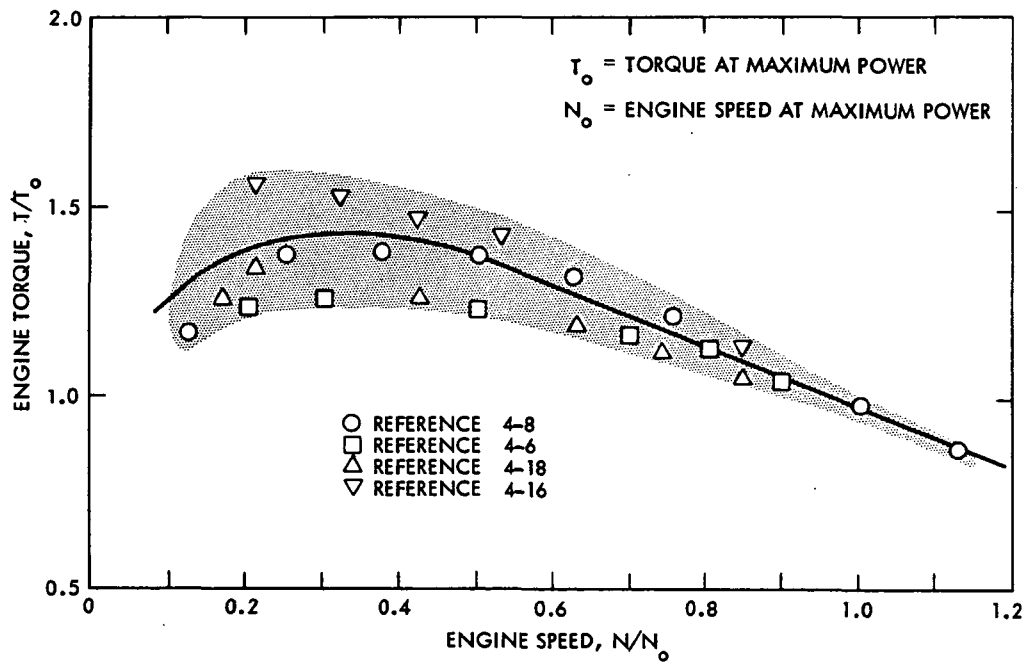


Figure 4-13. Torque-Speed Characteristics for Stirling

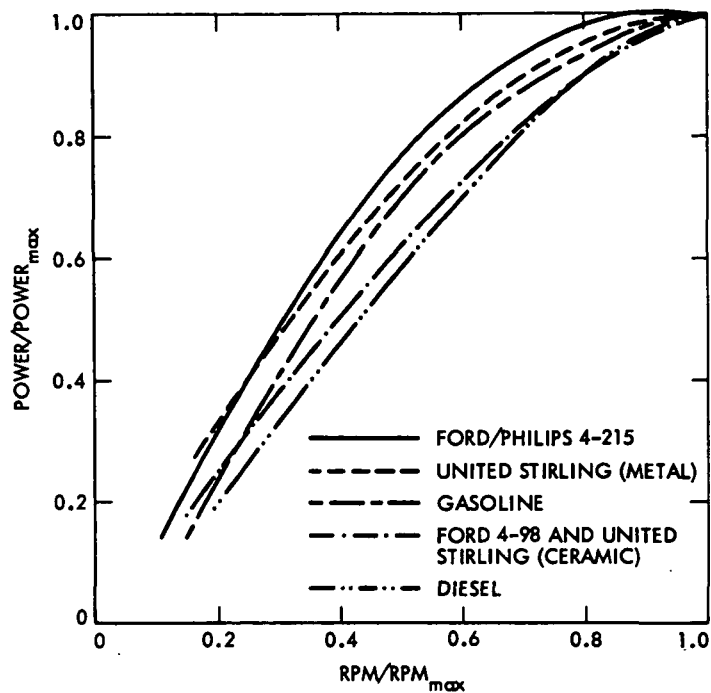


Figure 4-14. Power Limit Curves for Stirling and Conventional Engines

Table 4-4. Total Accessory Loads for Stirling and I.C. Gasoline Engines

Engine RPM	Total Accessory Load (% of Rated Power)	
	Stirling	I.C. Gasoline
1000	1.9	1.5
1500	2.8	--
2000	4.3	3.0
2500	6.6	--
3000	10.0	5.4
4000	18.0	--

4.4.3.4 Efficiency and Fuel Consumption. The primary reason for the current interest in the Stirling engine is the high inherent thermal efficiency of the cycle. If its associated ideal cycle is considered, its efficiency is equal to that of a Carnot engine operating between the same temperature limits. Various aspects of determining the overall or effective thermal efficiency of real Stirling engines, starting with the reference Carnot efficiency of the ideal engine, are discussed in Reference 4-16. If the approach given in Reference 4-16 is followed, the effective efficiency of the Stirling engine can be written as

$$\eta_{\text{eff}} = P_{\text{net}}/E_F$$

where

P_{net} = net shaft power with all auxiliaries driven

E_F = fuel energy flow

It is convenient to write η_{eff} in terms of a number of part efficiencies as

$$\eta_{\text{eff}} = \eta_H \eta_I \eta_M f_A$$

where

η_H = efficiency of the external heating system

η_I = indicated cycle efficiency

η_M = mechanical efficiency of the drive system

f_A = auxiliaries power factor

The indicated cycle efficiency η_I is related to the associated Carnot efficiency η_C by

$$\eta_I = C \eta_C$$

where

$$\eta_C = 1 - T_{\text{min}}/T_{\text{max}} = 1 - T_C/T_H$$

T_H = maximum working fluid temperature (°K)

T_C = coolant temperature (°K)

C = Carnot coefficient for the real engine

Hence

$$\eta_{\text{eff}} = C \left(1 - \frac{T_C}{T_H}\right) \eta_H \eta_M f_A$$

For automotive applications, it is estimated in Reference 4-16 that for well-designed engines

$$\eta_C = 0.65 - 0.75$$

$$\eta_H = 0.85 - 0.90$$

$$\eta_M = 0.85 - 0.90$$

$$f_A = 0.85 - 0.90$$

Thus

$$\eta_{\text{eff}} = K_S \left(1 - T_C/T_H\right)$$

where

$$0.4 < K_S < 0.55$$

Using this equation, the efficiency of Stirling engines can be estimated for various maximum working fluid and coolant temperatures. Typical results for a coolant temperature of 70°C are shown in the following table.

T_H (°C)	Efficiency (%)	
	Upper	Lower
700	35.5	26
800	37.5	27
1000	40	29
1200	42	31

The relationship between η_{eff} and bsfc is

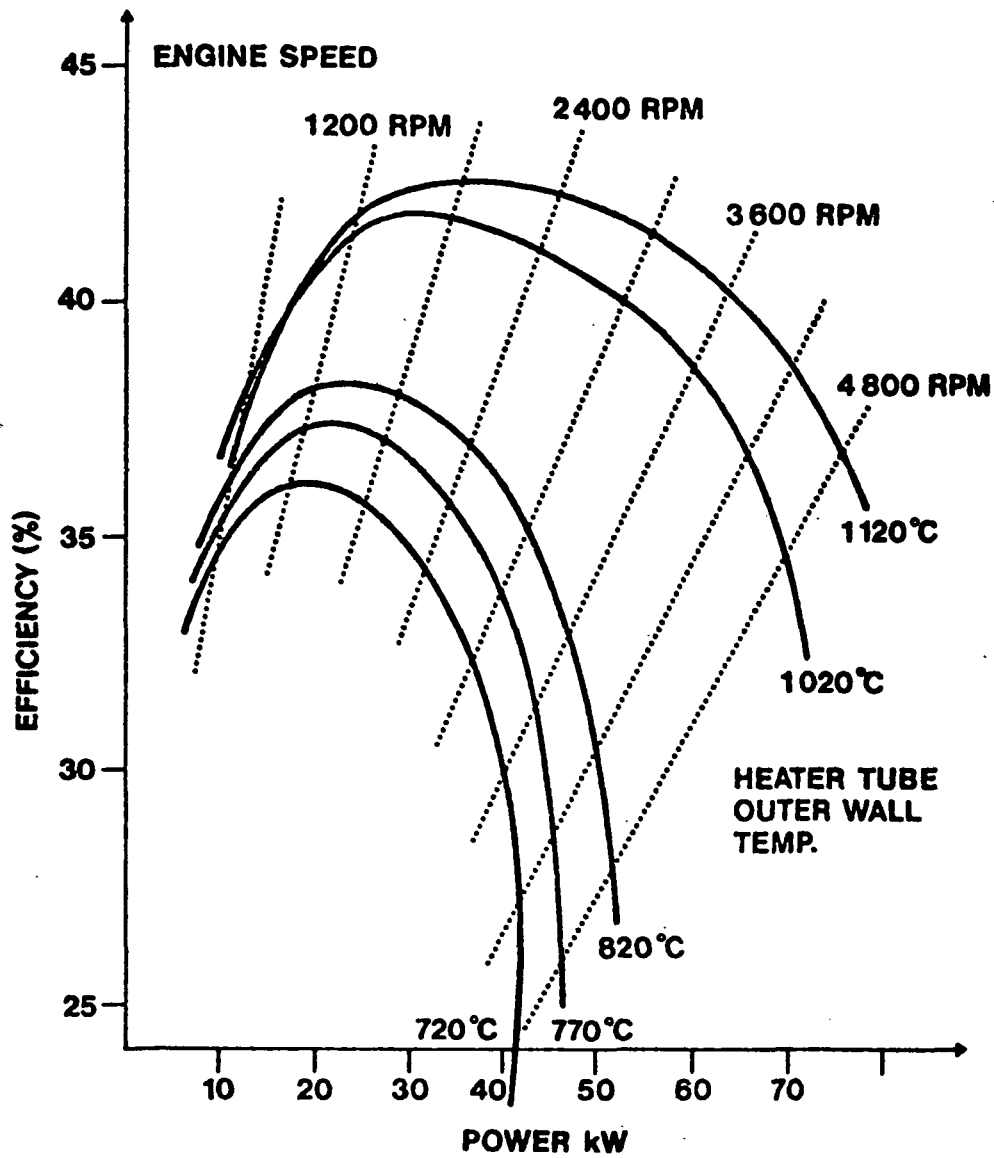
$$\text{bsfc (lb/bhp-h)} = 0.136/\eta_{\text{eff}}$$

for a fuel having an energy density of 18,600 Btu/lb. For a Stirling engine, the effect of a higher coolant temperature is to reduce efficiency. The sensitivity is about 0.5% for each 10°C change in coolant temperature. Calculated efficiency results for Stirling engines having heater tube temperatures between 720°C and 1120°C are shown in Figure 4-15, which was taken from Reference 4-16.

Fuel consumption maps (bsfc vs. power and engine RPM) are available in the literature for the Ford/Philips and United Stirling engines discussed in Section 4.4.2. The only engine for which there is a complete fuel consumption map based entirely on test data is the Ford/Philips 4-215. Some test data is also available for the United Stirling P40 and P75 engines. A compilation of available Stirling engine fuel consumption information (both test data and engineering calculations) is given in Figures 4-16 through 4-20. The data was used to prepare the generic tables of Stirling engine fuel consumption (bsfc vs % power and %RPM) presented in Tables 4-5 through 4-8. Those tables were used in obtaining the vehicle fuel economy computer results discussed in Section 4.4.4.4.

4.4.3.5 Emissions. Relatively little information is available concerning emissions (HC, CO, NO_x) from Stirling engines. This situation exists because it is felt by both Ford and United Stirling that attaining low emissions (i.e., meeting the 0.4 g/mi HC, 3.4 g/mi CO, 0.4 g/mi NO_x emissions levels) with the Stirling engine will not be difficult because it is an external combustion engine. For this reason, other R&D problems have had higher priority.

Ford has obtained some emissions data on the Ford/Philips 4-215 engine. Some of that data is shown in Table 4-9 at selected engine operating points. Since the engine is operated at lean air/fuel (A/F) ratios (18-22), the HC and CO emissions are quite low and vehicle chassis dynamometer tests have confirmed that the EPA urban driving cycle emissions are acceptable except possibly for some cold-start conditions. Attaining low NO_x emissions has been more difficult and has required the use of large amounts of exhaust gas recirculation (EGR) to reduce the peak temperatures in the burner. As indicated in Table 4-9, rates of exhaust gas recirculation vary from 75 to 100 percent depending on the operating point. The limited data on Stirling engine emissions seems to indicate that the low emissions goals can be met, but that more work is needed to further reduce NO_x emissions and HC and CO emissions during cold starts.



COOLING WATER TEMPERATURE 70°C

HEATER TUBE OUTER WALL TEMP. °C	CORRESPONDING WORKING GAS TEMP. °C	EXTERNAL HEATING SYSTEM EFFICIENCY %
720	650	0.90
770	700	0.89
820	750	0.88
1020	950	0.85
1120	1050	0.84

Figure 4-15. Calculated Effective Efficiency of Stirling Engines as a Function of Heater Head Temperature and Engine RPM

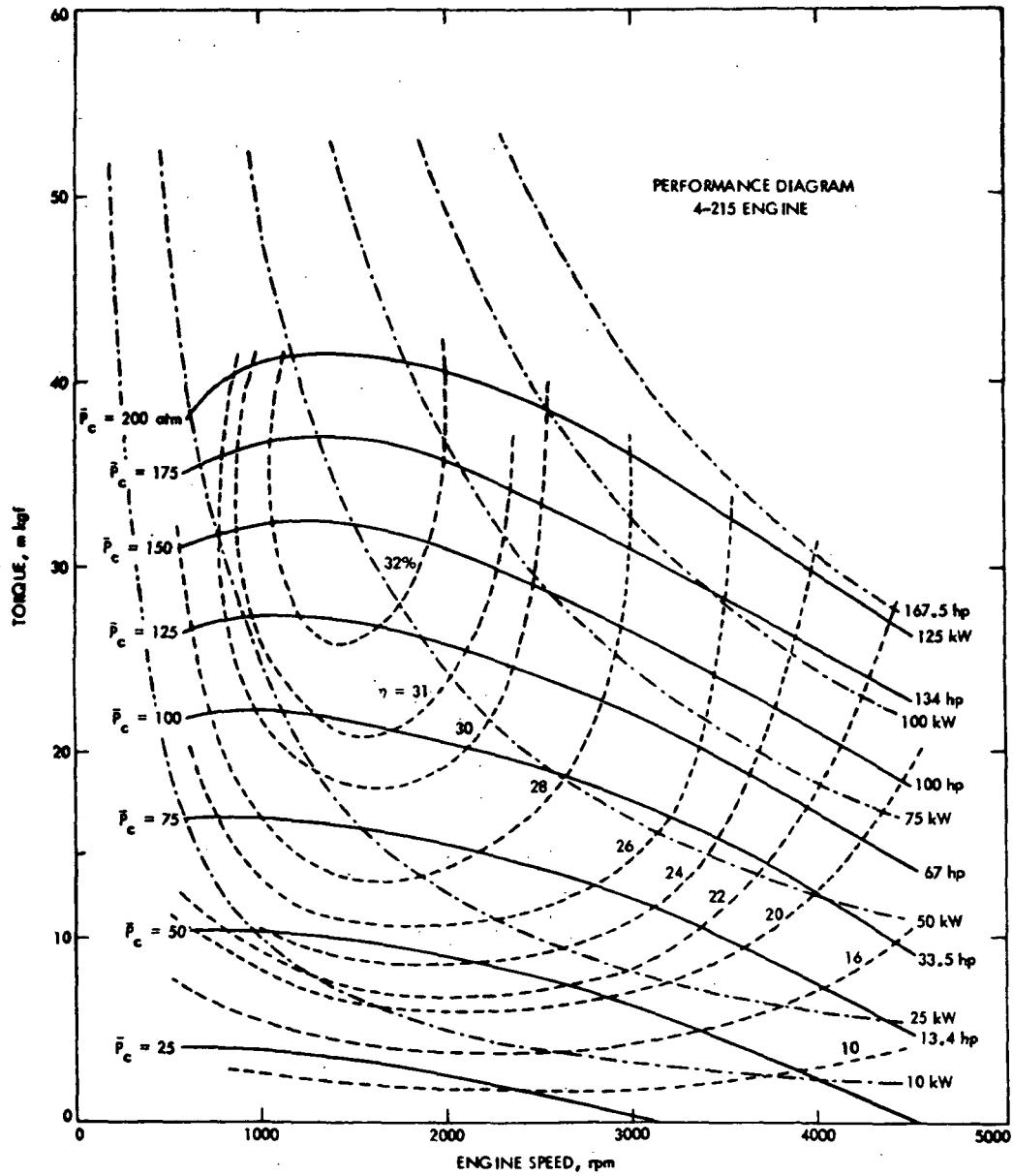


Figure 4-16. Performance Map for the Ford/Philips 4-215 Engine

MEAN PRESSURE POWER CONTROL

RADIATOR FAN DRIVEN, PERFORMANCE RELATED TO
AMBIENT AIR TEMP = 30 °C AND ALTITUDE = 150 m

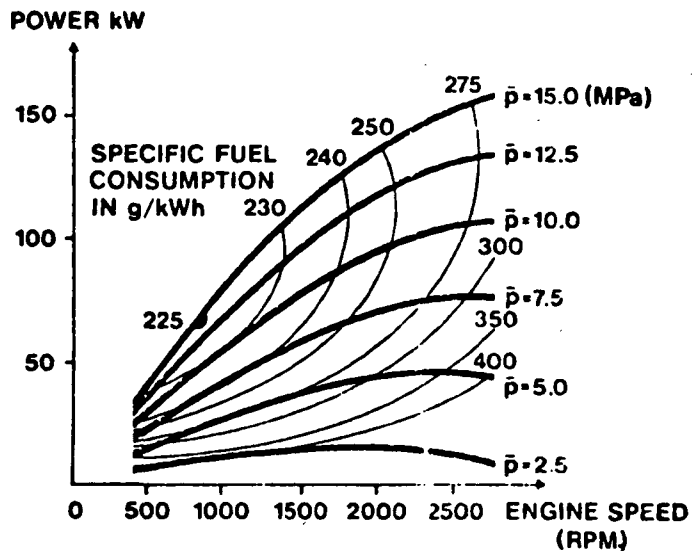


Figure 4-17. Performance Map for the United Stirling P150 Engine

4.4.4 Characteristics of Stirling Engine-Powered Vehicles

4.4.4.1 Vehicle Programs. As indicated in Table 4-10, there have been relatively few programs involved with the installation and testing of Stirling engines in vehicles. Very limited fuel economy data are available from those programs; in fact, fuel economy data is not currently available for a single engine/vehicle combination from both road and chassis dynamometer tests. One of the objectives of the Ford/Philips program with the Philips 4-215 engine in a Torino was to obtain such data, but frequent mechanical problems with the engine precluded it. United Stirling started road testing of the P75 engine in a Volvo distribution truck in the fall of 1977. Information from these tests is not available yet.

4.4.4.2 Packaging. Both Ford and United Stirling have shown that a Stirling engine can be installed (see Figures 4-21 and 4-22) in the engine compartment of a vehicle of conventional design, i.e., a rear-wheel-driven vehicle with an engine compartment sized to accommodate conventional L4, L6, and V-8 engines. In the Ford study of Reference 4-6, it was concluded that the 4-98 engine could be mounted for front-wheel drive in the Pinto, but the engine compartment of the Pinto is probably larger than that of a downsized vehicle having the same interior

PERFORMANCE MAP **4-98 STIRLING ENGINE WITH ROLLSOCK SEALS**

NET TORQUE VS ENGINE SPEED
 RADIATOR TOP WATER TEMPERATURE = 50° C
 HEATER INSIDE WALL TEMPERATURE = 750° C

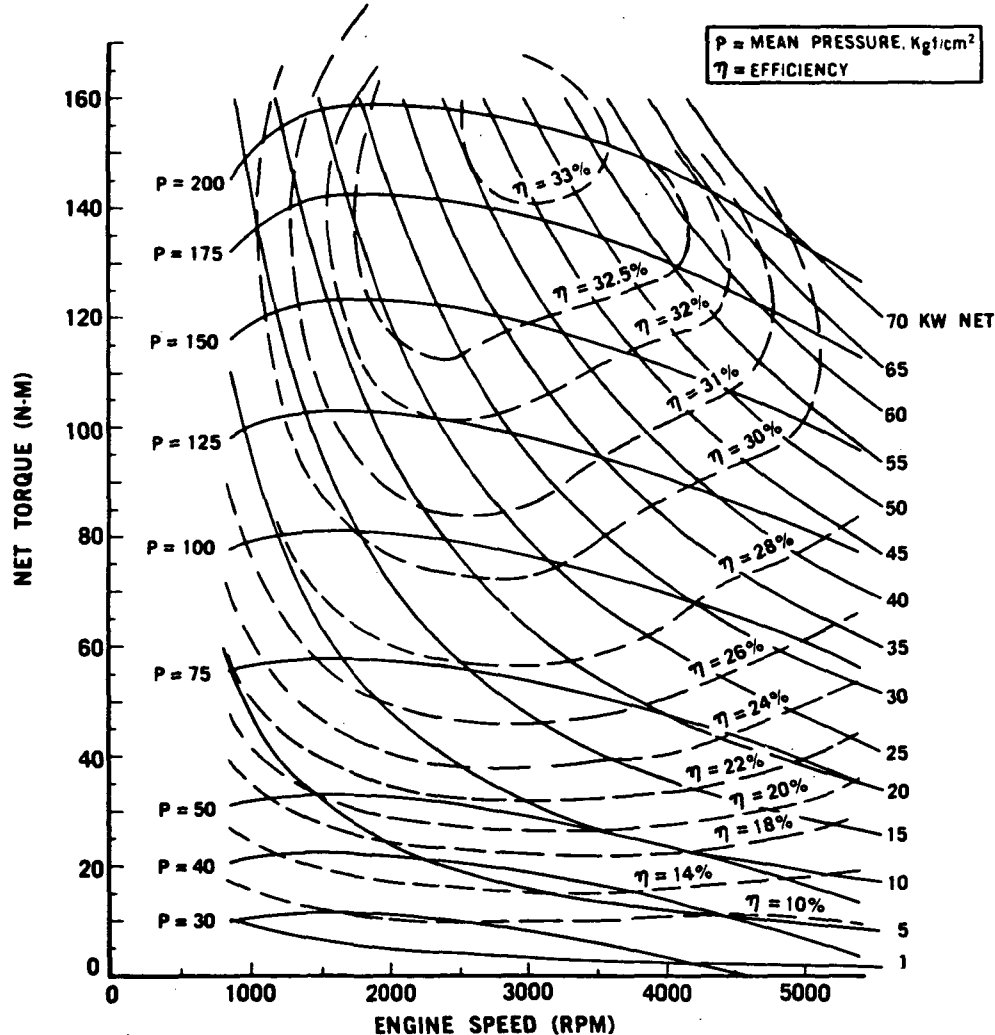


Figure 4-18. Performance Map for the Ford Stirling 4-98 Engine

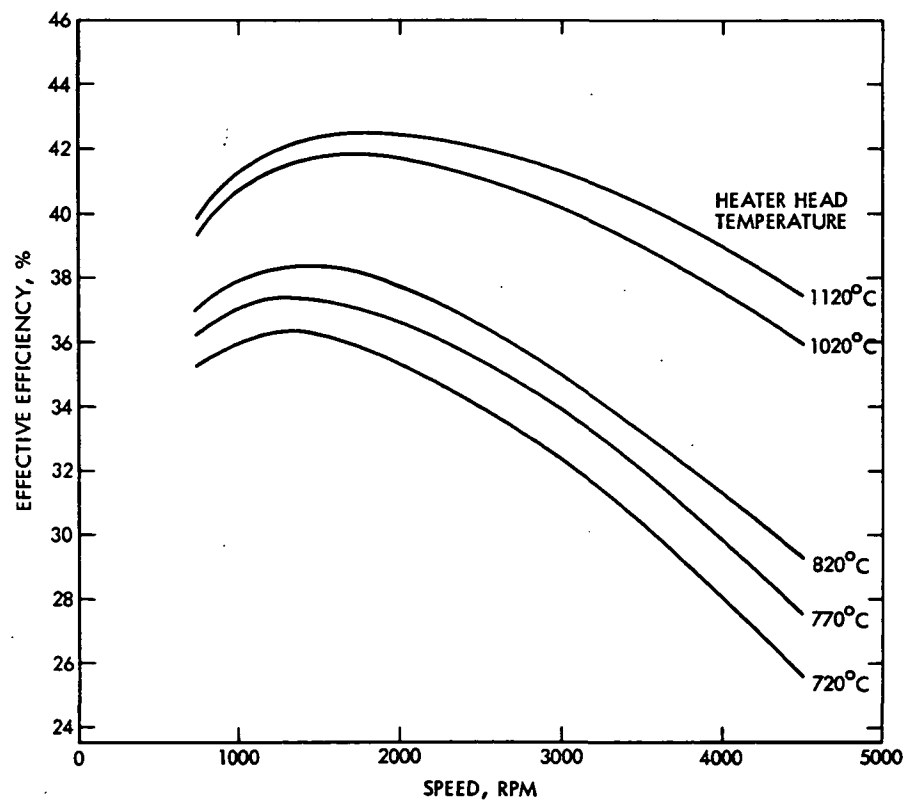
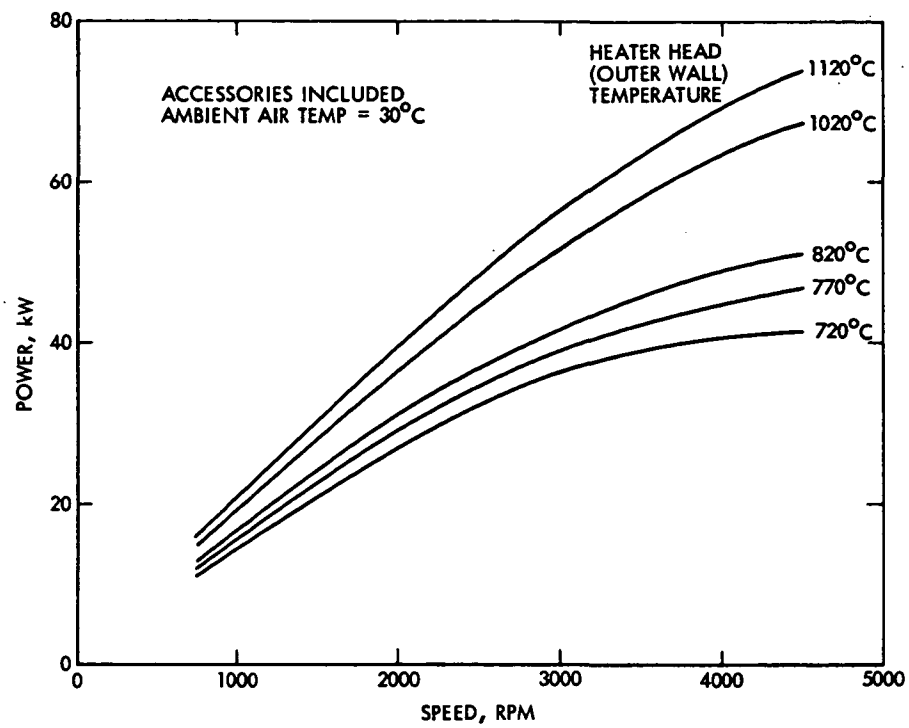


Figure 4-19. Performance Characteristics for the United Stirling P40 Engine

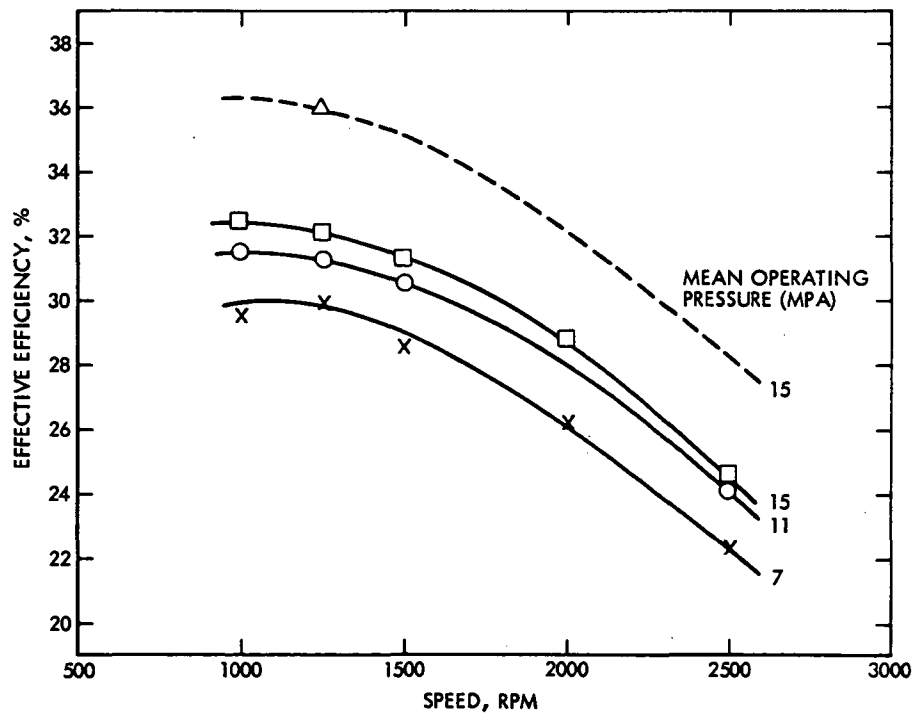
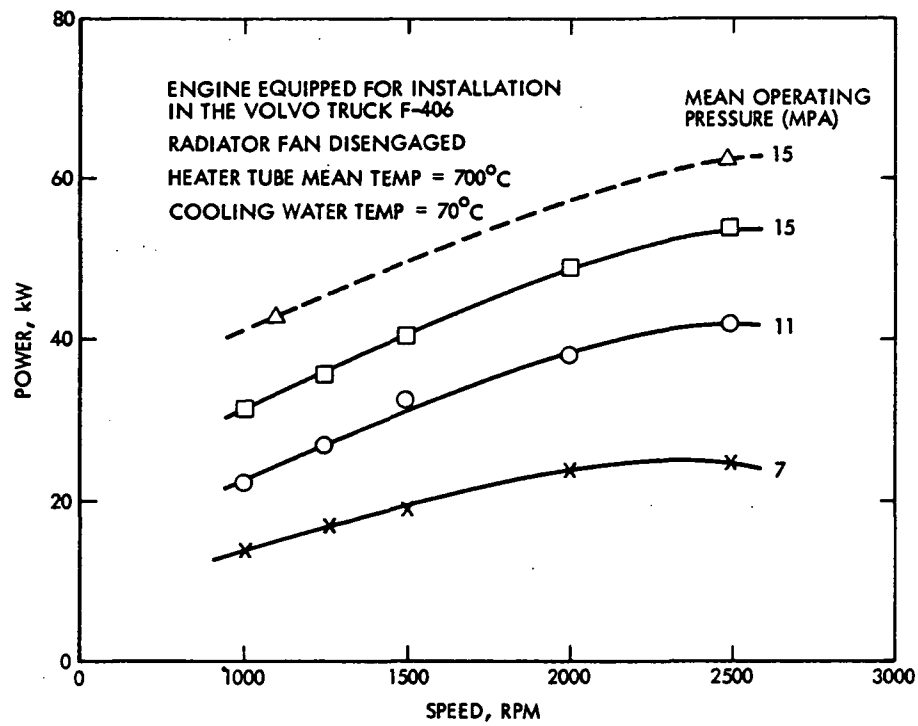


Figure 4-20. Performance Characteristics for the United Stirling P75 Engine

Table 4-5. Fuel Consumption and Power Limit Curve Data
for Ford/Philips 4-215 Stirling Engine

BRAKE SPECIFIC FUEL CONSUMPTION MAP DATA*									
f_{RPM}	f_{HP}								
	0.10	0.15	0.25	0.40	0.60	0.80	1.00		
0.111	1.490	1.080	0.838	0.609	0.532	0.500	0.489		
0.222	1.030	0.789	0.583	0.479	0.428	0.421	0.427		
0.333	0.893	0.670	0.532	0.459	0.421	0.419	0.419		
0.444	0.838	0.670	0.528	0.462	0.432	0.421	0.420		
0.555	0.838	0.713	0.558	0.489	0.459	0.447	0.447		
0.666	0.985	0.779	0.609	0.515	0.496	0.482	0.479		
0.777	1.080	0.839	0.698	0.588	0.545	0.519	0.515		
0.888	1.280	1.030	0.838	0.684	0.615	0.583	0.563		
1.000	1.680	1.340	1.200	0.865	0.744	0.670	0.644		
* $RPM_{idle} = 500$, $HP_{max} = 160$, $RPM_{max} = 4500$, idle fuel flow = 4.1 lb/h									
POWER LIMIT CURVE DATA									
f_{HP}	0.146	0.344	0.522	0.689	0.820	0.910	0.975	1.000	0.996
f_{RPM}	0.111	0.222	0.333	0.444	0.555	0.666	0.777	0.888	1.000
$f_{HP} = HP/HP_{max@rpm}$, $f_{RPM} = RPM/RPM_{max}$									

Table 4-6. Fuel Consumption and Power Limit Curve Data
for Ford 4-98 Stirling Engine

BRAKE SPECIFIC FUEL CONSUMPTION MAP DATA*										
f_{RPM}	f_{HP}									
	0.10	0.15	0.25	0.40	0.60	0.80	1.00			
0.15	1.910	1.190	0.788	0.609	0.515	0.462	0.432			
0.20	1.340	0.957	0.670	0.545	0.470	0.425	0.412			
0.30	1.060	0.744	0.571	0.479	0.419	0.399	0.394			
0.40	0.893	0.705	0.545	0.454	0.412	0.394	0.394			
0.50	0.859	0.694	0.536	0.451	0.412	0.394	0.394			
0.60	0.882	0.694	0.540	0.454	0.416	0.400	0.394			
0.70	0.893	0.698	0.558	0.486	0.425	0.406	0.398			
0.80	0.905	0.713	0.588	0.493	0.444	0.419	0.404			
0.90	0.957	0.744	0.620	0.515	0.465	0.442	0.416			
1.00	1.010	0.893	0.705	0.570	0.493	0.456	0.439			
* $RPM_{idle} = 750$, $HP_{max} = 82$, $RPM_{max} = 5000$, idle fuel flow = 2.5 lb/h										
POWER LIMIT CURVE DATA										
f_{HP}	0.180	0.247	0.379	0.504	0.620	0.728	0.817	0.896	0.955	1.000
f_{RPM}	0.15	0.20	0.30	0.40	0.50	0.60	0.70	0.80	0.90	1.00
$f_{HP} = HP/HP_{max@rpm}$, $f_{RPM} = RPM/RPM_{max}$										

Table 4-7. Fuel Consumption and Power Limit Curve Data
for United Stirling Metal Engine

BRAKE SPECIFIC FUEL CONSUMPTION MAP DATA*									
f_{RPM}	f_{HP}								
	0.05	0.20	0.35	0.50	0.65	0.80	1.00		
0.168	0.620	0.530	0.462	0.420	0.396	0.380	0.369		
0.263	0.562	0.435	0.380	0.368	0.360	0.356	0.350		
0.421	0.525	0.425	0.383	0.375	0.370	0.368	0.366		
0.579	0.530	0.460	0.413	0.396	0.391	0.389	0.386		
0.737	0.570	0.496	0.458	0.435	0.425	0.420	0.418		
0.895	0.650	0.577	0.522	0.475	0.472	0.467	0.466		
1.000	0.745	0.650	0.578	0.539	0.510	0.509	0.506		
* $RPM_{idle} = 800$, $HP_{max} = 120$, $RPM_{max} = 4750$, idle fuel flow = 2.5 lb/h									
POWER LIMIT CURVE DATA									
f_{HP}	0.330	0.479	0.617	0.745	0.840	0.915	0.957	1.000	1.000
f_{RPM}	0.210	0.316	0.421	0.526	0.632	0.737	0.842	0.947	1.000
$f_{HP} = HP/HP_{max@rpm}$, $f_{RPM} = RPM/RPM_{max}$									

Table 4-8. Fuel Consumption and Power Limit Curve Data
for United Stirling Ceramic Engine

BRAKE SPECIFIC FUEL CONSUMPTION MAP DATA*									
f_{RPM}	f_{HP}								
	0.05	0.20	0.35	0.50	0.65	0.80	1.00		
0.168	0.428	0.387	0.369	0.358	0.350	0.345	0.339		
0.263	0.372	0.354	0.342	0.335	0.330	0.327	0.328		
0.421	0.362	0.340	0.328	0.322	0.318	0.316	0.314		
0.579	0.365	0.345	0.336	0.331	0.326	0.322	0.316		
0.737	0.380	0.356	0.346	0.342	0.338	0.335	0.333		
0.895	0.528	0.402	0.368	0.355	0.349	0.346	0.344		
1.000	0.760	0.495	0.405	0.372	0.360	0.358	0.355		
* $RPM_{idle} = 800$, $HP_{max} = 120$, $RPM_{max} = 4750$, idle fuel flow = 2.5 lb/h									
POWER LIMIT CURVE DATA									
f_{HP}	0.210	0.280	0.400	0.530	0.640	0.750	0.830	0.900	1.000
f_{RPM}	0.168	0.210	0.315	0.421	0.526	0.631	0.737	0.842	1.000
$f_{HP} = HP/HP_{max@rpm}$, $f_{RPM} = RPM/RPM_{max}$									

Table 4-9. Stirling Engine Fuel Consumption and Emissions
Characteristics for the EPA Driving Cycles
(Ford/Phillips 4-215)

Simulated Point	% Time		RPM	HP	Fuel Consumption, lb/bhp-h	A/F	% EGR	Emissions, gm/(gm fuel x 10 ³)		
	Urban	Highway						HC	CO	NO _x
1	27.7	1.3	600	4.4	1.33	21	96	0.064	3.5	1.3
2	15.1	1.0	800	4.6	1.15	21	105	0.053	5.7	1.6
3	8.5	7.2	900	0.75	6.5	22	169	0.085	1.5	0.55
4	13.3	1.4	1000	9.2	0.85	21	136	0.073	4.4	1.1
5	14.4	1.7	1100	16.5	0.63	20	109	0.056	4.8	1.3
6	9.1	2.6	1300	27.9	0.56	20	94	0.051	6.9	1.9
7	2.9	4.8	1600	45.2	0.48	19	76	0.052	9.2	2.4
8	4.1	40.0	1700	25.7	0.59	18	91	0.062	9.1	1.4
9	2.0	5.2	1799	54	0.48	18	71	0.082	16.0	3.2
10	2.9	34.8	2000	33.9	0.56	18	82	0.075	8.1	2.2
Average over cycle								0.061	6.2	1.6

Table 4-10. Summary of Stirling Engine/Vehicle Programs

Year	Company	Engine	Vehicle	Type of Test	Status
1974	United Stirling	P40	Pinto	Road	Completed
1976	United Stirling	P40	Taunus station wagon	Road	Continuing
1976	Ford	Philips 4-215	Torino	Chassis dynamometer	Continuing
1977	United Stirling	P75	Volvo distribution truck	Road	Recently started
1978	United Stirling/MTI	P40	Opel	Road and chassis dynamometer	Recently started

dimensions. It seems likely that the design of the Stirling engine and the packaging of it into a vehicle will have to be considered together with the total vehicle system. Vehicles using Stirling engines are likely to be slightly larger and heavier than those using conventional Otto engines because of the larger size and weight of the Stirling engine. In this respect, the Stirling engine suffers a disadvantage similar to that of the naturally-aspirated diesel.

4.4.4.3 Performance and Driveability. Both Ford and United Stirling have demonstrated in their vehicle test programs that vehicles powered with Stirling engines can have satisfactory acceleration performance and driveability. As noted previously, the steady-state torque curve of the Stirling engine is well-suited for passenger car applications. Hence the primary question regarding the responsiveness of vehicles using a Stirling engine concerns the time required to increase the pressure of the hydrogen working fluid during a full-throttle acceleration. The experience of Ford with the Torino seems to indicate that the control approaches developed for blower/combustor and hydrogen systems work quite well and give good engine response.

One of the more difficult problems is the Stirling engine start-up (or key-on to drive away) time. This is always a point of concern with an external combustion engine, since the public has experienced the convenience of the fast starting of the conventional Otto engine. Ford has set a program goal of achieving a start-up time of

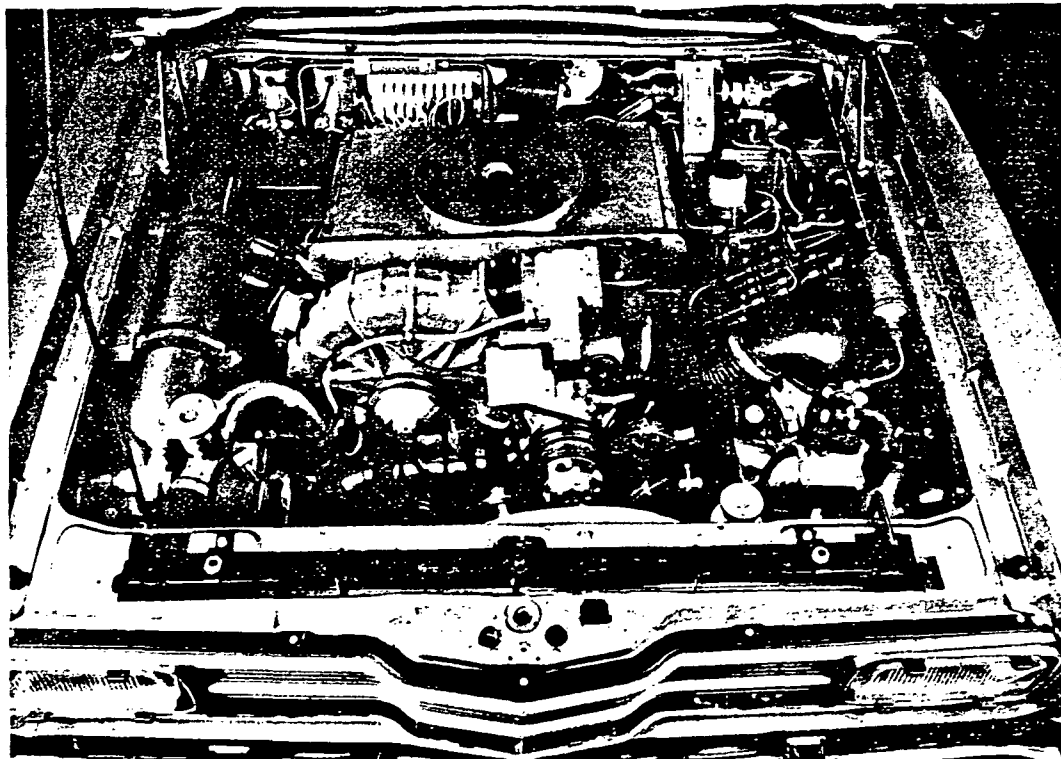


Figure 4-21. Ford Taunus Station Wagon with (bottom)
Stirling V4X35 Engine Installed

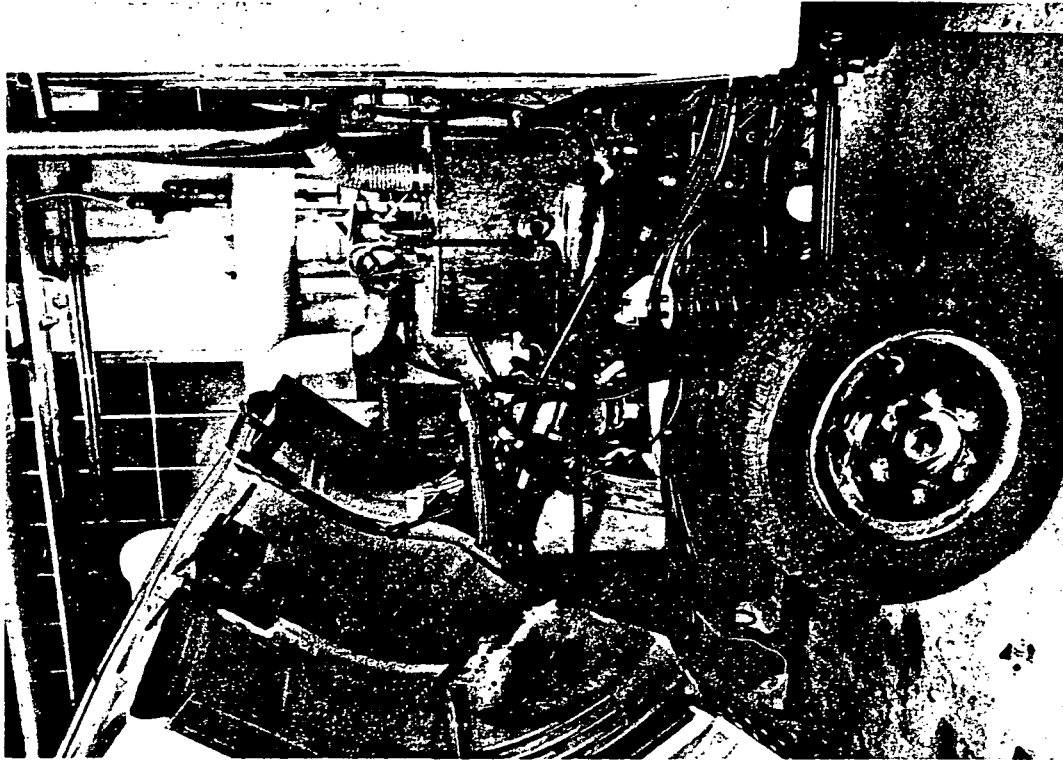
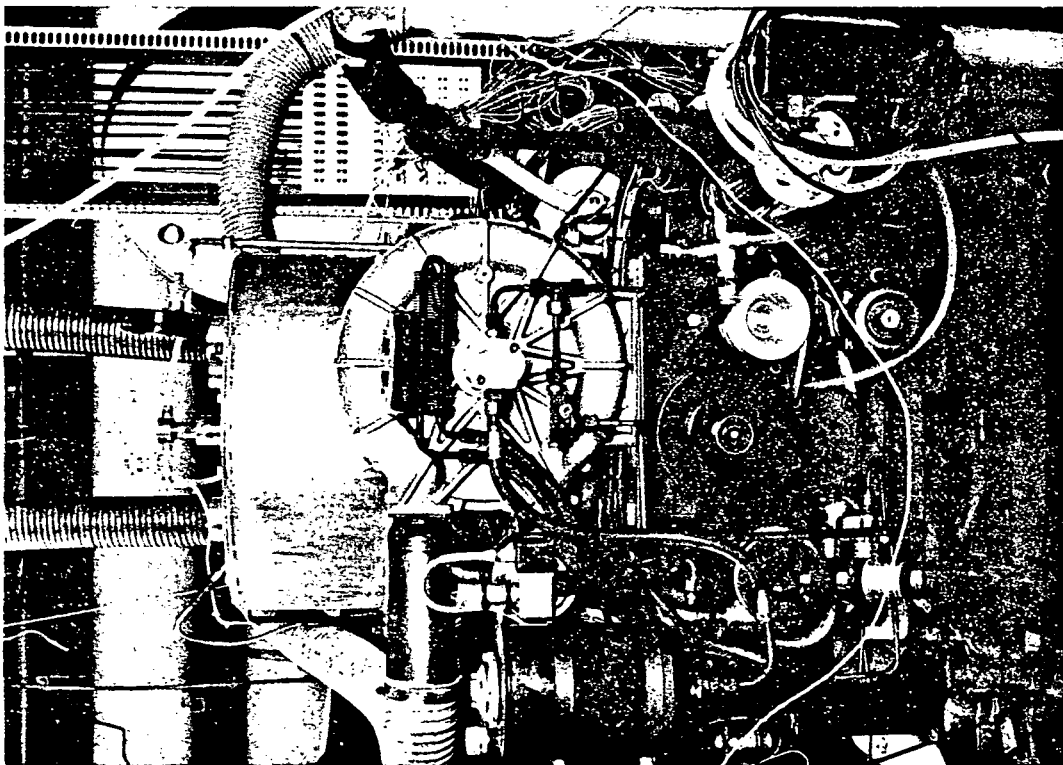


Figure 4-22. (left) Stirling P150 V4 Engine Module with Auxiliaries; (right) Stirling P150 Installed in a 13-Ton Medium Duty Delivery Truck

15 seconds; however, that goal has not been met yet. The Torino/4-215 engine has a measured start-up time of about 25 seconds at the present time.

4.4.4.4 Fuel Economy. The fuel economy of vehicles using Stirling engines is of particular interest and various aspects of that subject are discussed in this section. Since the data on the fuel economy of Stirling engine-equipped vehicles is very limited, most of the fuel economy information given in this section is based on computer vehicle simulations done at JPL using the Stirling engine bsfc maps shown in Tables 4-5 through 4-8.

A comparison of the JPL computer results and Ford test data for the Philips 4-215 engine in the Torino is given in Table 4-11. The computer results compare best with the Ford data obtained using the engine dynamometer approach. The computer predictions are closer for the highway cycle than for the urban cycle. In the latter, transient effects, such as losses in the power control system of the Stirling engine, are more important. Discussions with engineers at N.V. Philips indicated that throttling losses in the hydrogen pressure control system could reduce fuel economy by 10-15 percent from that based completely on

Table 4-11. Comparison of JPL Computer Simulation and Ford Test Data for the Philips 4-215 Engine in the Torino

Basis ⁽¹⁾	HP	City	Highway	Composite
JPL Computer Simulation	170	13.1	21.0	15.8
	135	14.1	24.0	17.3
Ford - engine map	170	14.0	17.5	15.3
Ford-chassis dynamometer (hot start)	170	12.5	20.6	15.2

(1) $I_w = 4500 \text{ lb}$

$C_D A = 11.0$

M4 transmission - JPL computer simulation

A3 transmission - Ford test

steady-state engine map data. In addition, no attempt was made to account for cold-start or warm-up effects in the JPL vehicle simulations. Additional work is clearly needed to bring the computer predictions of Stirling engine fuel economy into better agreement with the test data, but it appears that computer results are reasonable enough to permit their use at least on a comparison basis. Current results (May 1978) of the Ford fuel economy assessment activity (part of their DOE Stirling engine program) indicate that the fuel economy potential of an improved Stirling engine in a 4500 lb vehicle is 23.7 mpg for the composite driving cycle at a 40 percent confidence level (Reference 4-23).

Calculations of vehicle fuel economy were made for a number of Stirling engine designs and vehicle inertia weights. The results are summarized in Table 4-12. Based on comparisons of the computer results with the limited test data available, it is expected that the computer projections of fuel economy are optimistic rather than pessimistic. This is based on the fact that many of the engine bsfc maps used were calculated (by the engine developers) rather than being based on dynamometer data. Engine transient effects and associated losses were neglected in the computer simulation. In addition, it should be noted that for the most part, manual four- and five-speed transmissions, which have relatively high efficiencies, were used for calculations. This was done to circumvent difficulties in matching automatic transmissions to various engine/vehicle combinations and to avoid questions concerning the efficiencies of advanced automatic transmissions with lockup or power splitting.

The fuel economy potential of vehicles powered by Stirling engines is compared with that of vehicles powered by conventional and advanced engines in Section 9.

4.4.4.5 Emissions. The only emissions data available for Stirling engine-powered cars was taken by Ford on the Torino equipped with the Philips 4-215 engine. The limited data obtained are summarized in Table 4-13. The emissions (g/mi) measured were quite low, as is typical of an external combustion engine. Although the emissions of the Torino did not meet the program goals (0.4 g/mi HC, 3.4 g/mi CO, 0.4 g/mi NO_x), there is no reason to believe the emissions goals could not be met after limited additional effort. Meeting the 0.4 g/mi NO_x requirement with sufficient margin of safety is likely to be the most difficult emissions task, but additional work is also required in the area of cold-start HC emissions. All work to date indicates that automotive Stirling engines can be developed which have very clean exhausts without aftertreatment and that low emissions continues to be a major advantage of the engine.

Table 4-12. Summary of Computer Results for the
Fuel Economy of Vehicles Using
Stirling Engines

Engine	TRM	Iw	C_D^A	HP	Fuel Economy, mpg		
					U	H	Comp
Ford 4-98	M5	3500	10.9	120	21.0	31.1	24.6
Ford 4-98	CVT	3500	10.9	120	24.7	37.8	29.3
United Stirling (metal)	M5	3500	10.9	120	25.0	41.6	30.5
United Stirling (ceramic)	M5	3500	10.9	120	27.9	50.8	35.0
Ford 4-98	M5	2250	9.0	70	27.2	45.2	33.1
Ford 4-98	M5	2335	9.2	56	30.4	44.6	35.1
United Stirling (metal)	M4	2050	8.5	60	39.0	49.3	43.0
United Stirling (ceramic)	M4	2050	8.5	60	44.2	64.0	51.3
Philips 4-215	M4	4500	11.0	170	13.0	21.0	15.7
Philips 4-215	M4	4500	11.0	135	14.1	24.0	17.3

Table 4-13. Summary of Ford Torino Vehicle
Test Results Using the Philips
4-215 Engine

Test Procedure	Emissions, g/mi			Fuel Economy, mpg	
	HC	CO	NO _x	Urban	Highway
TESTS - JULY 1976; INERTIA WEIGHT = 4500 lb					
Engine Mapping					
Weighted Steady State ⁽¹⁾	0.17	1.27	0.54	13.0	16.9
Hot Start - CVS ⁽²⁾	0.20	1.53	0.60	13.0	16.9
Cold Start - CVS ⁽²⁾	0.59	1.99	0.66	11.7	16.9
Vehicle Chassis Dynamometer					
Hot Start	0.07	3.6	0.48	11.2	16.9
Cold Start	0.58	2.9	0.56	10.3	17.2
TEST - MARCH 1977; INERTIA WEIGHT = 5000 lb					
Engine Mapping					
Weighted Steady State ⁽¹⁾	0.01	1.27	0.32	14.0	17.5
Hot Start - CVS ⁽²⁾	0.01	1.52	0.35	14.0	17.5
Cold Start - CVS ⁽²⁾	0.04	1.98	0.39	12.6	17.5

Table 4-13. Summary of Ford Torino Vehicle
Test Results Using the Philips
4-215 Engine (Continuation 1)

Test Procedure	Emissions, g/mi			Fuel Economy, mpg	
	HC	CO	NO _x	Urban	Highway
Vehicle Chassis Dynamometer					
Hot Start	0.095	3.7	0.63	11.3	18.6

- (1) Steady-state engine dynamometer data weighted by fraction of time spent at selected operating points during EPA test cycles.
- (2) Engine mapping data point corrected by factor to account for average differences between engine dynamometer and vehicle chassis dynamometer tests.

4.5 CRITICAL PROBLEMS

The development of Stirling engines for the automotive application has been hampered by numerous mechanical problems which have prevented significant testing at the vehicle level. Also, the limited vehicle tests which have been run have not demonstrated the good fuel economy which is projected from engine test data. Beyond the solution of these immediate problems, it is necessary that cost-effective fabrication techniques and production methods be developed for Stirling engines if they are to be a viable alternative power plant for the future.

It is important that engine components be made more reliable and that losses which contribute to lower-than-expected vehicle fuel economy be reduced. At this early stage of development, the piston rod seal system has led to numerous engine failures. The roll-sock seal is easily damaged by contamination, difficult to install, and requires a complex oil support system. Failure of the roll-sock seal, which can result from contamination or failure of the oil support system, leads to a catastrophic engine failure requiring a complete engine teardown to remove oil from the working space. Significant improvements have been made in the sliding seal as an alternative to the roll-sock seal; however, hydrogen leakage in the sliding seal is still too large.

Current mean-pressure-level (MPL) power control systems have experienced control instabilities and have produced losses which significantly penalize vehicle fuel economy. The losses are a result of the short-circuiting method which is used to rapidly reduce engine power and the large compressor work needed as a result of leakage. These losses become more important during transient operations, such as those encountered on the urban driving cycle.

Significant progress has been made in the development of the ceramic rotary air preheater; however, leakage, seal wear, and durability continue to be problems. Although it has a lower effectiveness, the more conventional metallic recuperator-type of air preheater is still being developed since it solves many of the problems of the rotary air preheater.

Accessory loads for Stirling engines are much larger than for conventional gasoline and diesel engines. The largest parasitic loss is that due to the blower for the combustor. These large parasitic losses result in a substantial fuel economy penalty.

The heater head remains the most difficult design and fabrication problem in the Stirling engine. United Stirling is continuing an effort to develop simpler heater head designs which are easier to fabricate and repair. It is necessary that cost-effective fabrication techniques and production methods be developed for heater heads (metallic and ceramic) and other Stirling engine components.

4.6 RECOMMENDATIONS FOR FUTURE R&D

Based on an evaluation of the current status of Stirling engine technology, the following recommendations for future R&D activities are made:

- Continue development of independent (free from licensing agreements) performance and design models for Stirling engines. Develop simplified transient models for Stirling engine systems to aid in understanding the transient losses. This could minimize the penalty associated with the mean-pressure-level (MPL) power control systems.
- Develop the ceramics technology (material formulation, processing, fabrication, and ceramic/metallic joining techniques) needed to make hot-path components for an advanced Stirling engine. Pursue the development of a ceramic recuperator as an alternative to the ceramic rotary air preheater.
- Develop cost-effective fabrication techniques and production methods for Stirling engine components, especially the heater head.

- Develop a comprehensive design technique for the regenerator. Gather data on energy-storage heat exchangers for conditions existing in Stirling engines (i.e., hydrogen working fluid, high pressure, high cyclic frequency).
- Characterize the basic mechanisms for shaft seals and develop comprehensive design techniques for such seals. Characterize the mechanisms for heat transfer and aerodynamic losses for the oscillatory flow conditions which occur in each of the Stirling engine components, including heat exchangers and interconnecting passages. Develop comprehensive design techniques for all engine components.

4.7 CONCLUSIONS

The projected vehicle fuel economy (based on engine dynamometer test data) of Stirling-powered vehicles is significantly better than that of conventional vehicles; however, in limited vehicle tests this fuel economy advantage has not yet been demonstrated. This can be partially attributed to the losses in the mean-pressure-level (MPL) power control system. The losses are a result of the short-circuiting method which is used to rapidly reduce engine power and the large amount of compressor work needed as a result of leakage. These losses become more important during transient operations, such as those encountered on the urban driving cycle. Another significant penalty for Stirling engines arises from the fact that accessory loads (especially the combustor blower) are much larger than for conventional engines. Clearly, much additional work at the vehicle level is required to demonstrate the fuel economy potential of Stirling engines.

Progress has been made in achieving the good transient operation needed in the vehicle application. Driveability and engine response is adequate in some test systems.

The Stirling engine has advantages over the more conventional engines in the areas of multifuel capability and emissions/noise control. Stirling engines have the capability, with minor modifications, of using a broad variety of liquid fuels, gaseous fuels, and nonpetroleum fuels. The most stringent emissions standard (0.4 g/mi HC, 3.4 g/mi CO, 0.4 g/mi NO_x) can probably be met by Stirling engines in both large and small vehicles; however, this may require the use of more complex combustor designs, especially for the higher temperature advanced engines. Stirling engines also have a low noise capability compared with conventional engines.

As a result of the current vehicle downsizing trend, engine weight and size have become a more important consideration. At this stage of development it is not clear which Stirling engine configuration

(swashplate or crankshaft) is most advantageous for the automotive application. The swashplate design of the Ford/Philips 4-215 engine is smaller and lighter in weight, but has had limited durability. The crankshaft designs of United Stirling have been more durable, but are heavier and larger. In general, all current Stirling engines are larger and heavier than conventional gasoline and diesel engines of the same horsepower. Because of their more desirable torque-speed characteristics, Stirling engines should require a lower power-to-weight ratio than conventional engines for equivalent vehicle performance. In spite of this, packaging equivalent performance Stirling engines into downsized vehicles may be a problem, especially in engine compartments designed for modern, lightweight, (turbocharged) conventional engines.

At the component level, much additional development work is needed to simplify designs, reduce losses, improve reliability, and develop more cost-effective fabrication techniques and better production methods for Stirling engines. The heater head remains the most difficult design and fabrication problem in the Stirling engine. Although considerable progress has been made in the development of the ceramic rotary air preheater, some problems with leakage, seal wear, and durability remain. There is a need for a ceramic recuperator effort as an alternative approach. Progress has also been made in power control systems; however, control instabilities and thermal efficiency penalties continue to be problems with present systems. The piston rod seal continues to be a problem with both the roll-sock and sliding seals in current use. Although more appropriate for production, the sliding seal still permits too much hydrogen leakage at this stage of its development.

Since projections for future alternative engine systems (Stirling) are usually, of necessity, based on engine map data, it is desirable to make calculations for the baseline systems in a similar manner. Frequently, however, the required data are unavailable or company proprietary. Therefore, there is a need for more accurate engine and vehicle data (emissions and fuel consumption) for conventional gasoline and diesel baseline systems.

Notes to Section 4

1. Source: Fact-finding visit by ATSP team members to United Stirling of Sweden, Malmo, Sweden, Sept. 13, 1977.
2. Source: Fact-finding visit by ATSP team members to N. V. Philips Gloeilampenfabrieken, Eindhoven, The Netherlands, Sept. 19, 1977.
3. Source: Fact-finding visit by ATSP team members to Ford Motor Company, Nov. 17, 1976.

SECTION 5

ADVANCED TRANSMISSIONS

5.1 INTRODUCTION

In this section, current activities in the areas of advanced automotive transmissions are reviewed. The advanced transmissions discussed include (1) modified automatic transmissions utilizing torque converter lockup or torque-splitting and (2) continuously variable units (CVT) of both the hydromechanical and traction-drive types.

5.2 ADVANCED AUTOMATIC TRANSMISSIONS

5.2.1 Transmission Configurations and Present Status of Development

With the mandate to improve passenger car fuel economy, there has been considerable activity recently in the area of advanced automatic transmissions. Figure 5-1 shows schematically the various transmission configurations which are being studied, and for which, in most cases, hardware is being developed. All of the transmissions depicted are automatic in the sense that gear shifting and clutch operation are not controlled by the driver. The reference or baseline transmission is the three-speed automatic (A3) using a three-element torque converter and an automatic shifting three-speed gear box. The mechanical complexity of the transmissions increases progressively from the A3 to the CVT units with the addition of an overdrive gear, torque converter lockup clutch, torque split capability, and finally an infinitely variable unit of the traction-drive or hydraulic type. In all cases, the objective is to operate the engine at as low a bsfc as possible and to transfer the maximum fraction of the power through mechanical gears to the wheels.

The CVT is simply the end point of a logical progression beginning with the standard automatic transmission (A3). Whether or not the inclusion of an infinitely variable unit in the transmission is justified depends on economic, reliability, driveability, and fuel economy considerations. Sufficient information and data is not yet available to determine which of the advanced automatic transmissions would be the most cost-effective from either the automobile industry or consumer point-of-view.

The present status of development of the various advanced transmissions is summarized in Table 5-1. As indicated in the table the three-speed automatic transmission with torque converter lockup (Chrysler) and the four-speed, overdrive automatic transmission without torque converter lockup (Toyota) are presently being marketed in limited numbers by the automobile industry and a first generation, prototype

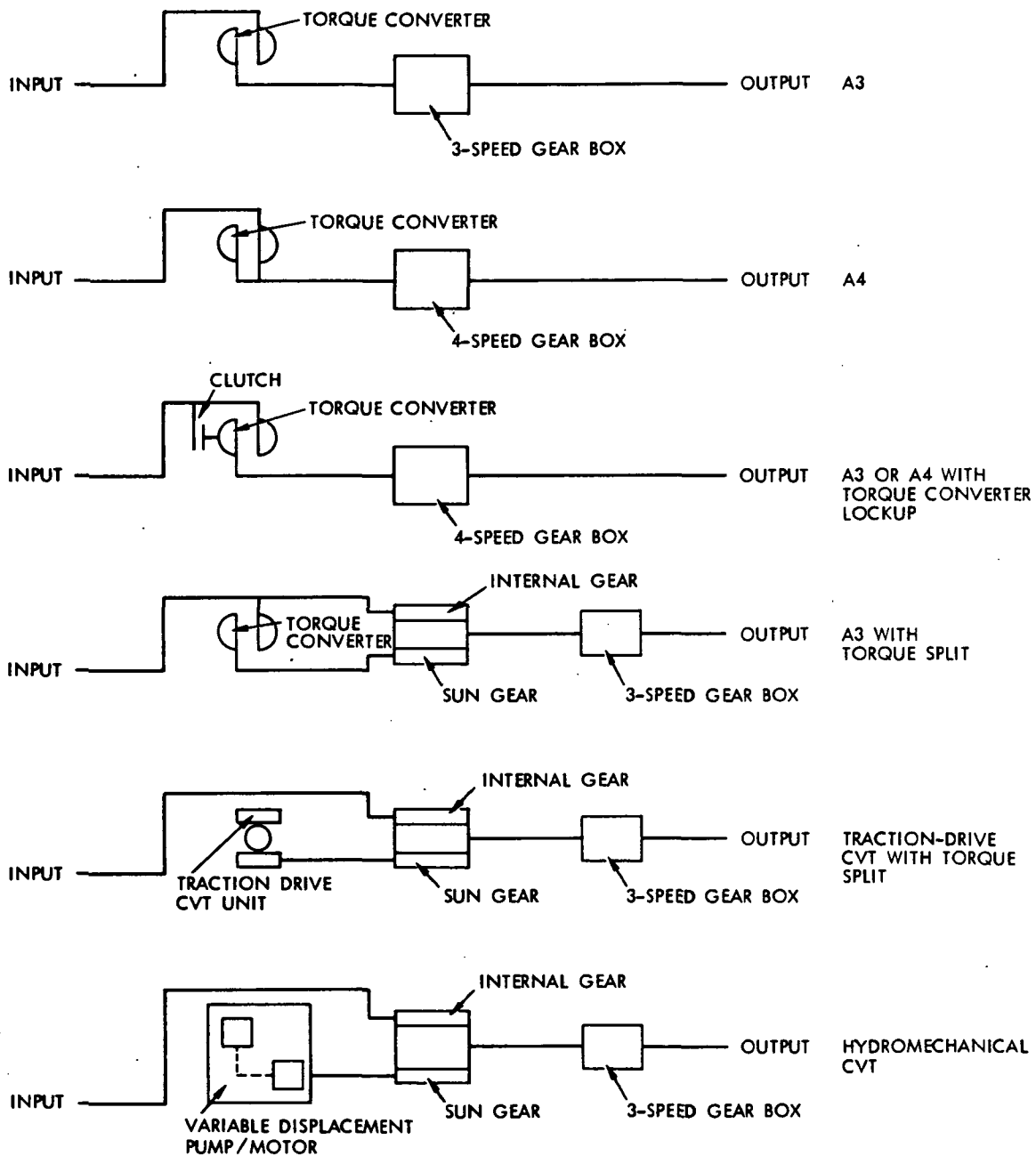


Figure 5-1. Advanced Automatic Transmission Configurations (Reference 5-1)

Table 5-1. Summary of Advanced Transmission Developments

Transmission Type	Special Feature	Investigator/ Developer	Status
Three-speed with lockup in third gear	Lockup clutch combined with standard three-speed automatic	Chrysler	Marketed in 1978
		General Motors	Under development
Four-speed with overdrive	Fourth over-drive gear combined with standard three-speed automatic	Toyota	Marketed in 1978
		Ford	Under development
Three-speed with torque split	-	Ford	Under development
Traction-drive CVT without torque split	Belt drive with variable pitch pulleys	DAF/Volvo	Marketed in 1977
	Flexible torus discs pressing against spherical rollers	Ford/DOE	Design study
Traction - drive CVT with torque split	Toroidal drive	Garrett/Navy	Design study and hardware development
Hydromechanical CVT with torque split	Variable displacement hydraulic pump/motor units	Orshansky/DOE	Vehicle testing of first generation unit and continuing R&D

hydromechanical CVT is being tested in an AMC Hornet by Orshansky under contract with the Department of Energy. This latter program is reviewed in some detail in Section 5.2.3. A less complex and efficient traction-drive CVT unit developed by DAF is used in the Volvo 343 subcompact car, which is marketed in Europe. The DAF CVT is not thought to be suitable for use in larger vehicles requiring significantly greater horsepower than the Volvo 343 or for applications requiring high efficiency over a wide speed range.

5.2.2 Projected Improvements in Fuel Economy Using Advanced Automatic Transmissions

There have been a number of studies (References 5-2 through 5-8) of vehicle fuel economy using various advanced automatic transmissions. Most of these studies involved computer simulation of vehicles over the EPA urban and highway driving cycles. Recently, limited fuel economy data has become available for comparison with the computer calculations. Table 5-2 presents a summary of simulation results of the fuel-economy effect of overdrive and torque converter lockup in automatic transmissions. The Chrysler and Toyota test data shown in the table indicate that the measured fuel-economy improvements with overdrive and lockup are less than those predicted in some instances. Simulation results obtained by several groups (References 5-5 through 5-8) for continuously variable transmissions are given in Tables 5-3 and 5-4. There is considerable variation in the predicted fuel economy improvement resulting from the use of a CVT rather than a standard three-speed automatic. Most of the variation from study-to-study is likely due to differences in the assumed CVT efficiency characteristics and engine operating line. The significant effect of the choice of the engine operating line on the fuel economy improvement obtained with the CVT is discussed in the next section in connection with the Orshansky hydromechanical CVT.

It is of interest to compare the fuel-economy improvements, relative to the standard three-speed automatic, that could be expected if the various advanced transmissions (depicted in Figure 5-1) were used. This is done in Table 5-5 based on the results given in Tables 5-2 to 5-4. The projected improvements shown in Table 5-5 are given in terms of ranges because the information currently available is simply not adequate to do otherwise. In general, it is seen in Table 5-5 that the magnitude of the improvement increases with the complexity of the transmission and that simply adding an overdrive gear improves the highway fuel economy by 15-20 percent.

There has been considerable discussion and disagreement in recent years concerning the relative advantages of continuously variable transmissions compared to more conventional automatics utilizing overdrive and torque converter lockup in one or more gears. Table 5-5 suggests that for highway driving the advantage of the CVT is quite small, but in urban driving it is significant - at least 10 percent. Comparisons should, of course, be made for equal vehicle performance in both acceleration and passing maneuvers. The CVT system has an additional degree of freedom not available to a transmission using a torque converter, but in order to translate the added capability to follow a prescribed engine operating line into higher fuel economy requires rather sophisticated control of transmission operation and maintenance of transmission efficiencies of 85-90 percent over a wide range of power and speed ratio. Whether or not the transmissions using either a torque converter or an infinitely variable unit will yield better fuel economy than the five-speed geared transmissions with an automatic

Table 5-2. Summary of Fuel Economy Simulation Results
of the Effect of Overdrive and Lockup
Using Automatic Transmissions

Vehicle Information	Transmission Information			0-60 mph, Acceleration Time, sec	Fuel Economy (mpg) Improvement (%)		
	Number of Gears	Overall Gear Ratio Range	Lockup in Which Gears	Overdrive Gear Ratio	Urban	Comp.	Ref.
GENERAL MOTORS							
	3	2.84 to 1	-	-	-	Base	5-6
	3	2.84 to 1	3rd	-	-	6.7	
	4	4.22 to 1	3rd, 4th	0.68 to 1	-	14.3	
UNIVERSITY OF WISCONSIN							
1973 vehicle 4800 lb, 400 CID engine	3	-	-	-	Base		5-4
	3	-	2nd, 3rd	-	10.0		
	4	-	-	0.8 to 1	4.0		
	4	-	2nd, 3rd, 4th	0.8 to 1	16.0		
1972 vehicle 3150 lb, 250 CID engine	3	-	-	-	Base		5-4
	3	-	2nd, 3rd	-	5.0		
	4	-	-	0.8 to 1	5.0		
	4	-	2nd, 3rd, 4th	0.8 to 1	12.0		

Table 5-2. Summary of Fuel Economy Simulation Results
of the Effect of Overdrive and Lockup
Using Automatic Transmissions
(Continuation 1)

Vehicle Information	Transmission Information			0-60 mph, Acceleration Time, sec	Fuel Economy (mpg) Improvement (%)		
	Number of Gears	Overall Gear Ratio Range	Lockup in Which Gears	Overdrive Gear Ratio	Urban	Comp.	Ref.
TOYOTA							
1978 vehicle 3500 lb, L4 2.6 liter engine	3	2.48 to 1	-	-	Base 4.6(2.5) ¹	Base 10.8(9.3) ¹	5-3
	4	3.6 to 1	-	0.689 to 1	-	-	
CHRYSLER							
1978 vehicle 4000 lb, V8 360 CID engine	3	2.45 to 1	-	-	Base 4(3.6) ^{1,2}	Base 4.7(5.0) ^{1,2}	5-2
	3	2.45 to 1	3rd	-	-	-	

(1) Test data

(2) Average of test data for two vehicles, 1978 EPA Certification Data

Table 5-3. Summary of University of Wisconsin Simulation Results
Using a Traction Drive CVT (Ref. 5-5)

Vehicle Description	Axle Ratio	Engine CID	Accel., 50-70 mph, sec	Fuel Economy, mpg			
				Urban Cycle	Steady Speeds		
					30 mph	40 mph	50 mph
Base car ⁽¹⁾ (4800 lb) with standard auto- matic transmission	2.75	430 ⁽¹⁾	5.1	12.5	20.4	20.6	19.4
Car with traction drive CVT ⁽³⁾ (4800 lb)	2.1	343 ⁽¹⁾	5.1	14.7	28.7	29.0	24.3
% Improvement over base car				18	41	41	25
Base car ⁽²⁾ (3150 lb) with standard auto- matic transmission	2.79	250 ⁽²⁾	11.8	12.6	23.0	21.9	20.2
Car with traction drive CVT ⁽³⁾ (3150 lb)	2.52	222 ⁽²⁾	11.8	16.2	30.4	28.9	23.4
% Improvement over base car				28	32	32	11

⁽¹⁾ V-8 engine without emission controls from the late 1960s.
⁽²⁾ L6 engine with emission controls to meet 1973 standards.
⁽³⁾ Speed ratio range 6.5:1; efficiency: 85% at 10% max. load, 89% at 25% max load, 92% at higher loads.

Table 5-4. Summary of Fuel Economy Simulation Results Comparing Continuously Variable and Standard Automatic Transmissions

Type	CVT Transmission	Efficiency, %	Vehicle Weight, lb	Fuel Economy Improvement Using CVT, %		
	Overall Gear Ratio Range			Urban	Highway	Composite
GENERAL MOTORS (1), (2)						
12 speed with lockup	6.00:1	95	-	-	-	21.6
12 speed with lockup	6.00:1	90	-	-	-	15.7
UNIVERSITY OF WISCONSIN(1)						
Traction drive	6.5:1	85-92	4800	18	33	23.1
Traction drive	6.5:1	85-92	3150	28	24	26.5
ORSHANSKY TRANSMISSION CO. (3), (4)						
Hydromechanical	7.0:1	80-90	3500	16.5	25	19.5

(1) Power train optimized to maximize fuel economy maintaining specified performance.

(2) Shift schedule selected gear which yielded minimum bsfc for each vehicle operating condition.

(3) Reference 5-8.

(4) Same engine and axle ratio used for CVT and automatic transmission simulations; vehicle tests indicated essentially the same acceleration performance.

- (1) Power train optimized to maximize fuel economy maintaining specified performance.
(2) Shift schedule selected gear which yielded minimum bsfc for each vehicle operating condition.
(3) Reference 5-8.
(4) Same engine and axle ratio used for CVT and automatic transmission simulations; vehicle tests indicated essentially the same acceleration performance.

Table 5-5. Summary of Projected Fuel Economy Improvements
Using Various Advanced Automatic Transmissions

Type	Transmission				Fuel Economy Improvement, %		
	No. Gears	Overall Gear Ratio Range	Lockup	Efficiency, %			
					Urban	Highway	Composite
Automatic	3	3.0 to 1	No	-	0(base)	0	0
Automatic	3	3.0 to 1	Yes	-	5-10	5-10	5-10
Automatic	4	4.0 to 1	No	-	3- 5	15-20	8-11
Automatic	4	4.0 to 1	Yes	-	10-15	20-25	14-19
Continuously Variable	-	7.0 to 1	-	80-90	15-20	20-25	17-22
Continuously Variable	-	7.0 to 1	-	85-90	18-22	25-30	21-25

clutch and microprocessor-controlled shifting is not yet clear (Reference 5-9). What is clear is that many automatic-type transmissions are currently being developed and that they are all quite complex and will all require sophisticated controls to meet fuel-economy improvement goals and at the same time preserve acceptable vehicle driveability.

5.2.3 Hydromechanical CVT Development

5.2.3.1 Hardware Development and Testing. Hydromechanical continuously variable transmissions are being developed and tested by the Orshansky Transmission Corporation for the Department of Energy. This program has progressed through the design, component development, and laboratory testing of complete CVT units to the present stage of chassis dynamometer and road testing of prototype transmissions in vehicles. Most of the vehicle testing has been done in a 1975 AMC Hornet, but some tests have also been run in a Honda Civic and a Chevrolet Nova. As discussed in the next section, considerable fuel economy and emissions data have been published for the CVT in the AMC Hornet. Unfortunately, similar data has not yet been published for the tests involving the Honda Civic and the Nova.

The transmission (Figure 5-2) being tested by Orshansky is a two-range unit having the characteristics shown in Figure 5-3. Note that the speed ratio (N_0/N_1) can be varied continuously from about 0.25 to 1.9, with the efficiency being in the 80-90% range. Detailed characterization (efficiency vs. input torque, input speed, and output speed) of the unit in both the low and high ranges is given in Reference 5-8 based on extensive laboratory tests. The present prototype CVT unit is considerably heavier (about 200 lb) than a standard three-speed automatic, but it is expected that redesign of the unit will reduce the weight penalty of the CVT to 10-15 percent.

A three-range hydromechanical CVT is currently being designed (Reference 5-10) by Orshansky for use in a 3500 lb vehicle. The additional range results in closer spacing of the mechanical gear ratios and requires less power to be transferred through the hydraulic path, thus improving the efficiency of the transmission. In addition, the three-range unit should be quieter than the two-range unit for the same reason.

5.2.3.2 Vehicle Test Results. The Orshansky two-range hydromechanical CVT has been tested extensively in an AMC Hornet on a chassis dynamometer. In addition, the CVT-equipped vehicle has been road-tested for about 10,000 miles including several cross-country trips (San Diego to Detroit). As discussed in subsequent paragraphs, there has been steady progress in improving the fuel-economy performance of the CVT-equipped vehicle as the control system for the CVT has been improved, permitting the use of more advantageous engine operating lines. Various operating lines which have been used in testing the AMC Hornet are shown in Figure 5-4.

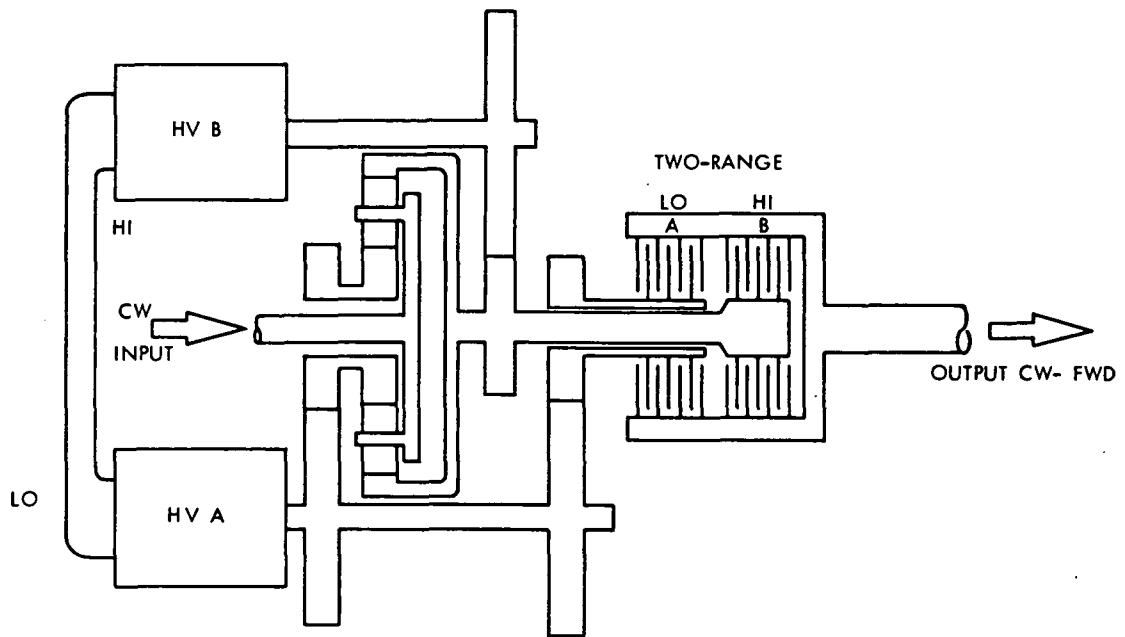


Figure 5-2. Orshansky Hydromechanical CVT

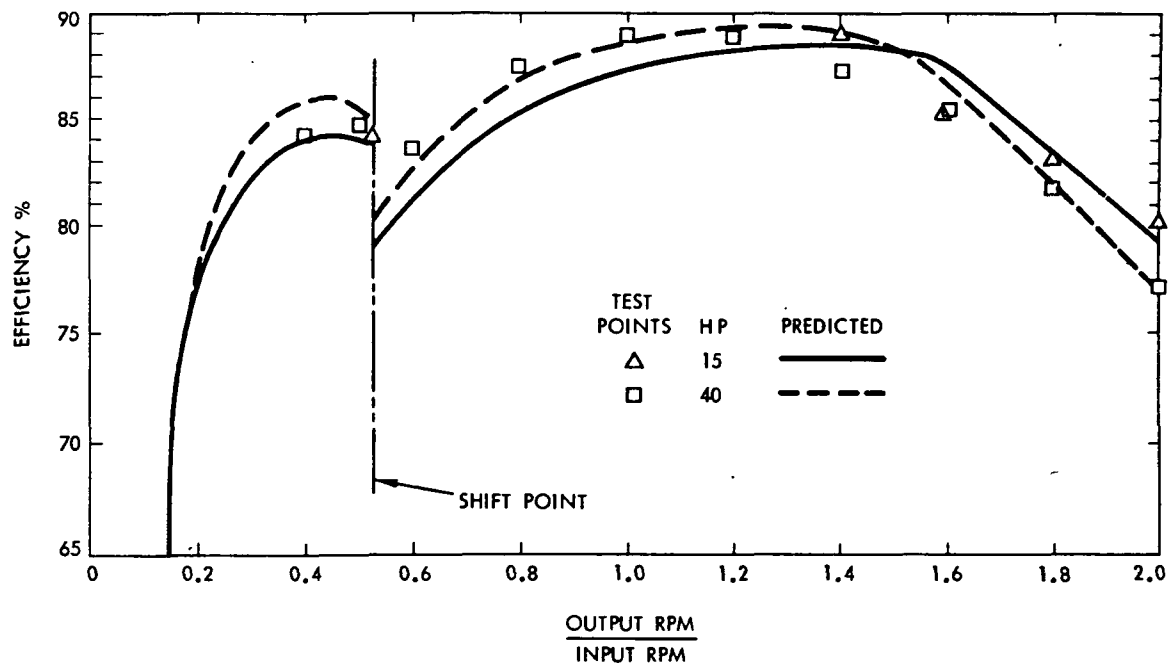


Figure 5-3. Two-Range Hydromechanical Transmission Efficiency, Test vs. Predicted

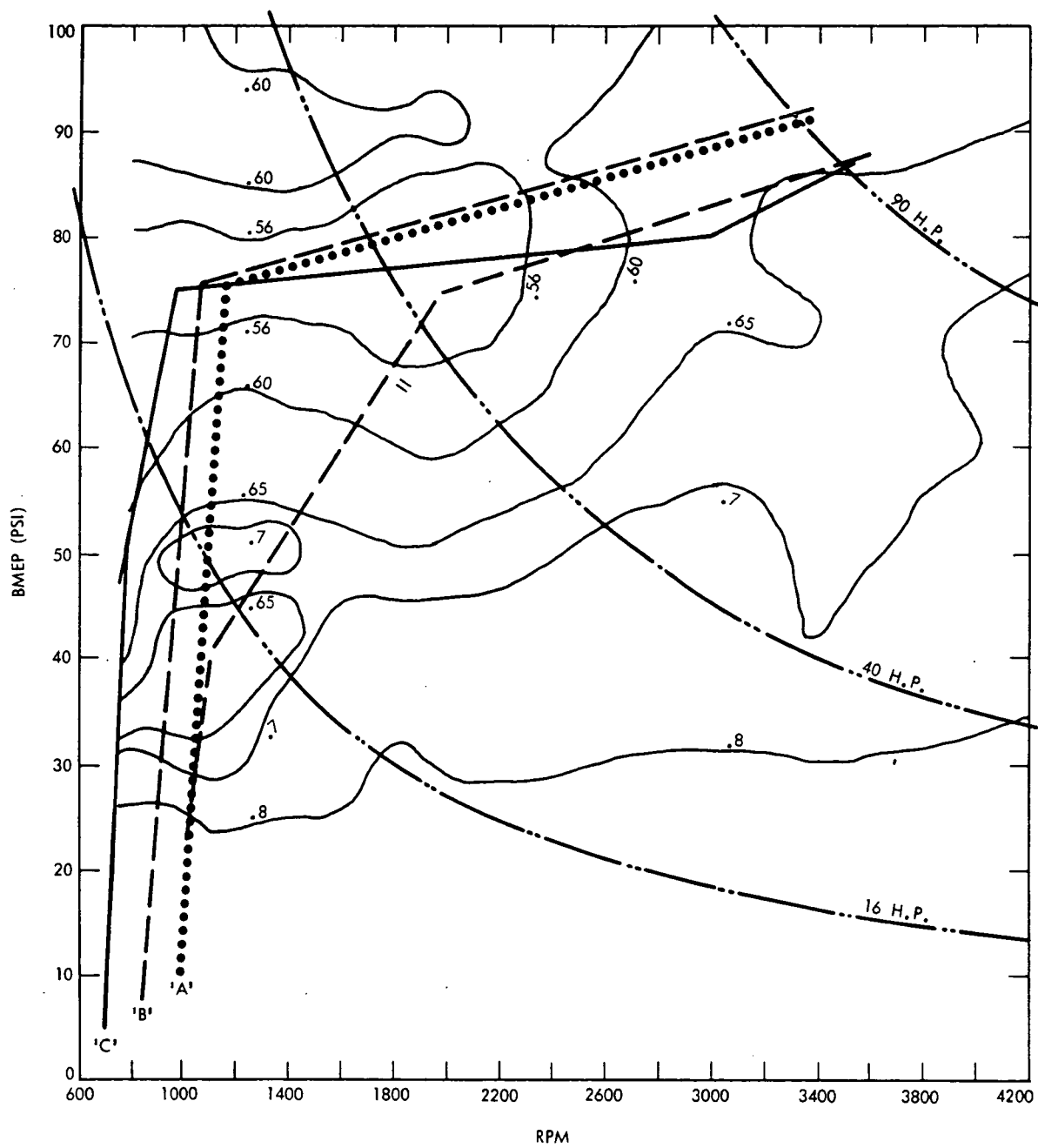


Figure 5-4. Engine Operating Schedule - AMC L6 232 CID

Constant-speed fuel-economy test data (References 5-8, 5-10, and 5-11) for the AMC Hornet have been obtained on the chassis dynamometer. A summary of such data is given in Table 5-6. The effect of engine operating line on fuel economy is clearly evident from a comparison of the results obtained using Schedules II and C. Fuel-economy data (References 5-8, 5-10, and 5-11) for the EPA urban and highway cycles are summarized in Table 5-7 for several engine operating lines. As in the case of constant speed driving, the effect of operating line is significant for driving over the urban and highway cycles. Hence it appears that, in order to attain the fuel economy improvement potential that is theoretically possible with the CVT, it will be necessary to develop control systems and engines which can sustain operation close to the minimum bsfc line (Schedule C in Figure 5-3). Comparisons between measured and predicted fuel economy results using the two-range CVT are also shown in Table 5-7. The comparisons show quite good validation of the computer simulations of CVT-equipped vehicles.

The emissions data given in Table 5-7 indicate all three regulated emissions (HC, CO, NO_x) are higher for vehicles using the CVT than the standard three-speed automatic. Whether or not this will prove to be a serious problem associated with the use of the CVT is difficult to assess at the present time because there has been no work to date in

Table 5-6. Summary of Constant-Speed Fuel-Economy Test Data for the AMC Hornet Using the Hydromechanical Transmission (CVT)

AMC Hornet L6, 232 CID, 1975 Vehicle Equipped to Meet Federal Emission Standards Iw = 3500 lb			
Speed, mph	Fuel Economy, mpg		
	HMT ⁽¹⁾		
	Sched. II	Sched. C	A ₃
20	25.5 (28.8) ⁽²⁾	33.0	30.3 (29.9) ⁽²⁾
30	33.8 (33.3)	41.5	31.0 (30.2)
40	36.7 (35.7)	40.9	28.8 (28.7)
50	25.5 (25.0)	38.2	23.6 (23.2)

(1) Hydromechanical transmission - two-range

(2) () = calculated values

References 5-8, 5-10, 5-11

Table 5-7. Summary of EPA Urban Cycle Fuel Economy and Emissions Test Data for the AMC Hornet Using the Hydromechanical Transmission (CVT)

AMC Hornet
L6, 232 CID, 1975 Vehicle Equipped to Meet
Federal Emissions Standards
Iw = 3500 lb

Transmission	Fuel Economy, mpg		Emissions, g/mi		
	Urban	Highway	HC	CO	NO _x
HMT ⁽¹⁾ - Sched B ⁽³⁾	18.7	26.2	1.0	12.7	2.5
HMT - Sched C ⁽³⁾	23.5	31.7	1.1	9.5	3.7
A ₃ - Standard	19.4	23.2	0.94	3.0	2.9
HMT - Sched II ⁽³⁾	18.6	26.6	1.5	15.9	3.4
A ₃ - Standard	16.0	21.1	1.1	5.5	2.5
HMT - Sched I ^(2, 3)	19.5	26.6	1.2	7.3	3.5
HMT - Sched I (calc.)	18.4	25.0	0.85	12.5	-
A ₃ - Standard ^(2, 3)	16.5	21.1	0.95	1.5	2.3
A ₃ - Standard (calc.)	15.8	20.0	0.67	15.2	-

(1) Hydromechanical transmission - two-range

(2) Test results, hot transient, and stabilized phases only in urban cycle.

(3) See Figure 5-3 for the engine operating Schedules B, C, II.

Refs. 5-8, 5-10, 5-11

which the CVT was used with an engine having advanced emission control systems (catalysts, programmed EGR, etc.). Recalibration of the engine from both the fuel consumption and emission points-of-view will undoubtedly be necessary to optimize its use with a continuously variable transmission. Thus far, the vehicle tests with the Orshansky CVT have used stock engines with little or no modification. The NO_x emission increase is most likely to be a continuing problem because the use of a CVT results in engine operation at high BMEP and low RPM where the specific NO_x emissions (g/bhp-h) tend to be high. However,

as is shown in Reference 5-12, high EGR flow rates can be used at those engine operating conditions to reduce NO_x with a minimal effect on bsfc. Detailed study of emissions from CVT-equipped vehicles is clearly needed.

5.2.3.3 Critical Problems. The critical problems associated with the further development of hydromechanical continuously variable transmissions are as follows:

- (1) Reduction of noise from the hydraulic units at high load.
- (2) Weight reduction.
- (3) Recalibration of engines to optimize their use with the CVT.
- (4) Emission control.
- (5) Control system development to permit engine operation close to the minimum bsfc line with acceptable driveability.

SECTION 6

ALTERNATIVE AUTOMOTIVE FUELS

6.1 FUEL PRODUCTION AND RESOURCES

There have been a number of recent studies (References 6-1 through 6-3) of alternative automotive fuels and alternative resources from which the fuels can be produced. These studies have considered in detail the processes which could be used to produce the various fuels and the ultimate cost of the fuel for use in transportation vehicles. The present study was not primarily concerned with the production and cost of the alternative fuels, but instead with how the fuel requirements of the various alternative automotive engines would affect the fuels which could be used in the transportation sector, particularly passenger cars. The effect of the fuel requirements on the overall energy-use efficiency - from the primary energy resource to vehicle wheels - was of special interest, especially as it relates to the use of coal, oil shale, or biomass as primary energy resources for transportation (References 6-4 through 6-6).

There are a large number of alternative fuels which could be considered for use in automotive engines, but only the following hydrocarbon fuels will be included in the present discussion: (1) gasoline, (2) diesel fuel, (3) alcohols (methanol and ethanol), and (4) wide-cut distillates. These fuels could be used alone or in blends with one another to fuel automotive engines. The wide-cut distillate fuels are, essentially, the primary products of the distillation process and have components with properties ranging between those of gasoline and diesel oil - that is, components with boiling points between 100 and 650°F, octane numbers between 40 and 100, and cetane numbers between 20 and 70. The properties of the various hydrocarbon fuels are summarized in Table 6-1. It is evident from the table that there are large differences in their physical and chemical characteristics which would be expected to affect their use in engines. That this is indeed the case is discussed in the next section.

All of the fuels of interest are currently produced from crude oil, natural gas, and/or agricultural products or waste and could be produced from coal or oil shale if desired using known, but not necessarily economically viable, processes. The process flow charts given in Figures 6-1 through 6-4 illustrate the complex character of the chemical conversion processes and the various options available in varying the output products of the refinery or conversion plant irrespective of whether the input energy resource is crude oil, coal, or oil shale. In assessing the attractiveness of a particular fuel, the energy efficiency of producing the fuel from the input energy resource is important. This factor has been considered in most of the alternative fuel studies, but there has been some confusion (References 6-4 through 6-6) concerning the proper interpretation of the energy conversion efficiency results as they relate to the optimum fuel mix from a refinery or coal/oil shale conversion plant. The confusion seems to arise from whether one

Table 6-1. Properties of Various Alternative Hydrocarbon Fuels for Vehicles

Property	Gasoline	Diesel Oil	Kerosene	Naphtha	Methanol	Ethanol
Density, lb/gal	6.0-6.4	6.7-7.3	6.7	6.4	6.7	6.5
Boiling range, °F	100-400	325-650	300-840	150-300	148	172
Heat of Vaporization, Btu/lb	130	155	-	145	473	370
Ignition temperature, °F	495	490	491	450-530	878	738
Lower heating value, Btu/lb	19,290	18,500	19,000	18,700	9,100	11,600
Stoichiometric ratio, lb air/lb fuel	14.9	14.2	14.9	15.0	6.5	9.0
Octane number						
RON	92-100	-	-	60-70	106	106
MON	84-92	-	-	50-60	92	89
Cetane number	18	40-68	40-65	-	-	-

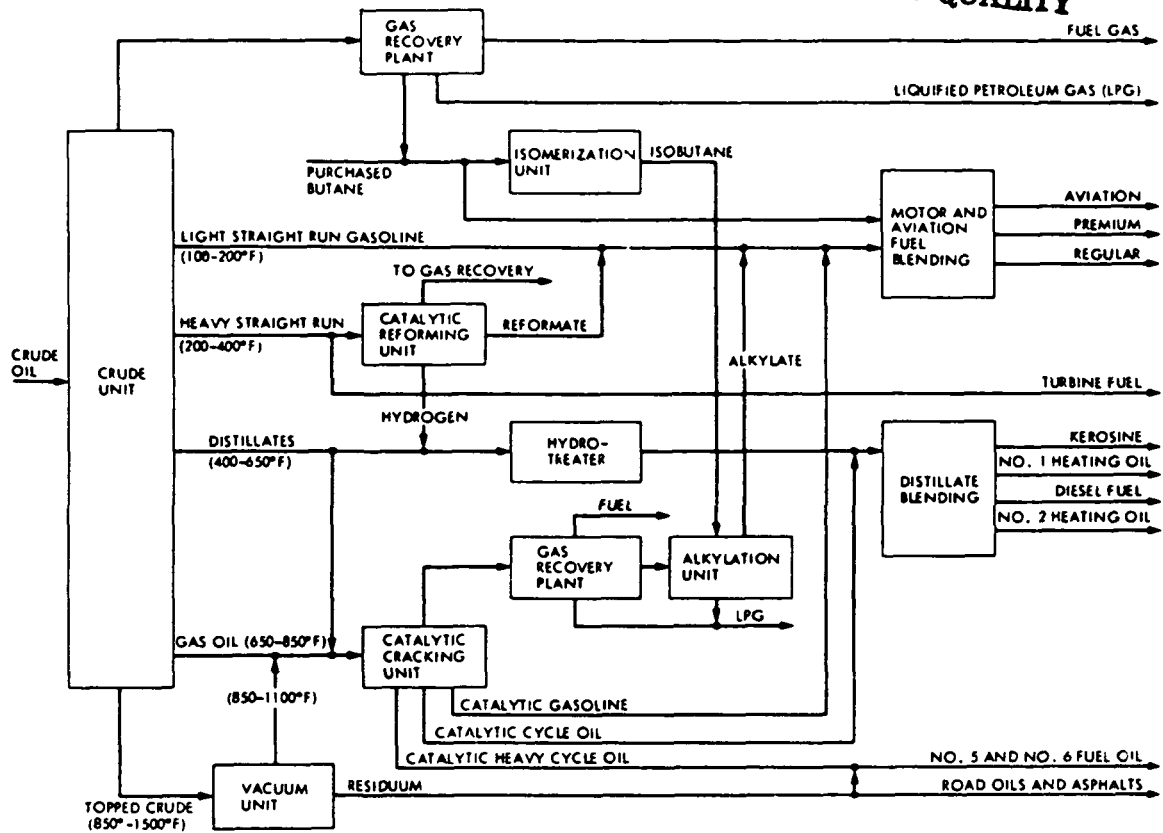


Figure 6-1. Sample Refinery (Reference 6-7)

considers the input energy resource-to-specific fuel conversion efficiency, which can be relatively low, or the input energy resource-to-total product conversion efficiency, which is usually considerably higher. Typical values for the conversion efficiencies for fuels obtained from crude oil, coal, and oil shale are given in Table 6-2 and Figure 6-5.

Information (References 6-9, 6-10) concerning changes in oil refinery process energy use for various product fuel mixes is given in Table 6-3. Even comparing extreme cases for which the primary products are unleaded gasoline and wide-cut distillate fuel, the change in energy use is only about 3.5%. The energy conversion efficiency information presented indicates that from a total energy-use point-of-view, the primary energy resource used to produce the fuel - that is, whether it is crude oil, coal, or oil shale - has a greater effect on the conversion efficiency than the fuel or fuels being produced. This appears to be especially true when fuels other than wide-cut distillates are required. Hence automotive engines which can use wide-cut distillates rather than the higher quality fuels such as gasoline, alcohols, and even diesel oil will be particularly attractive when crude oil is no longer the prime resource used to produce automotive fuels.

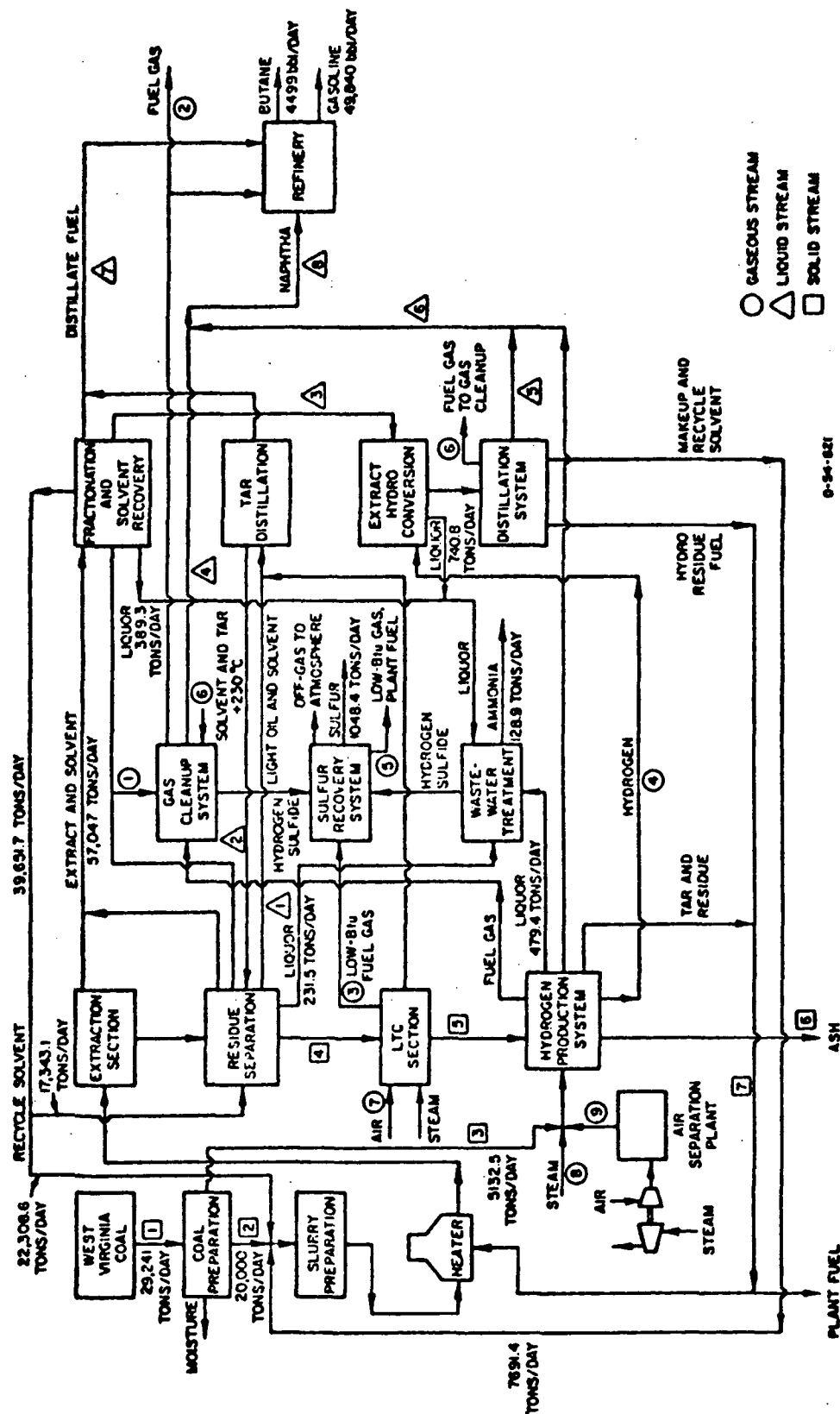
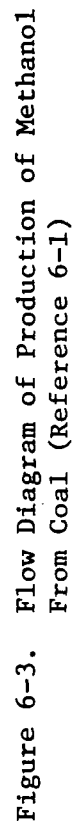


Figure 6-2. Flow Diagram of CSF - Process Production of Gasoline (50,000 bbl/day) from Coal (Reference 6-1)

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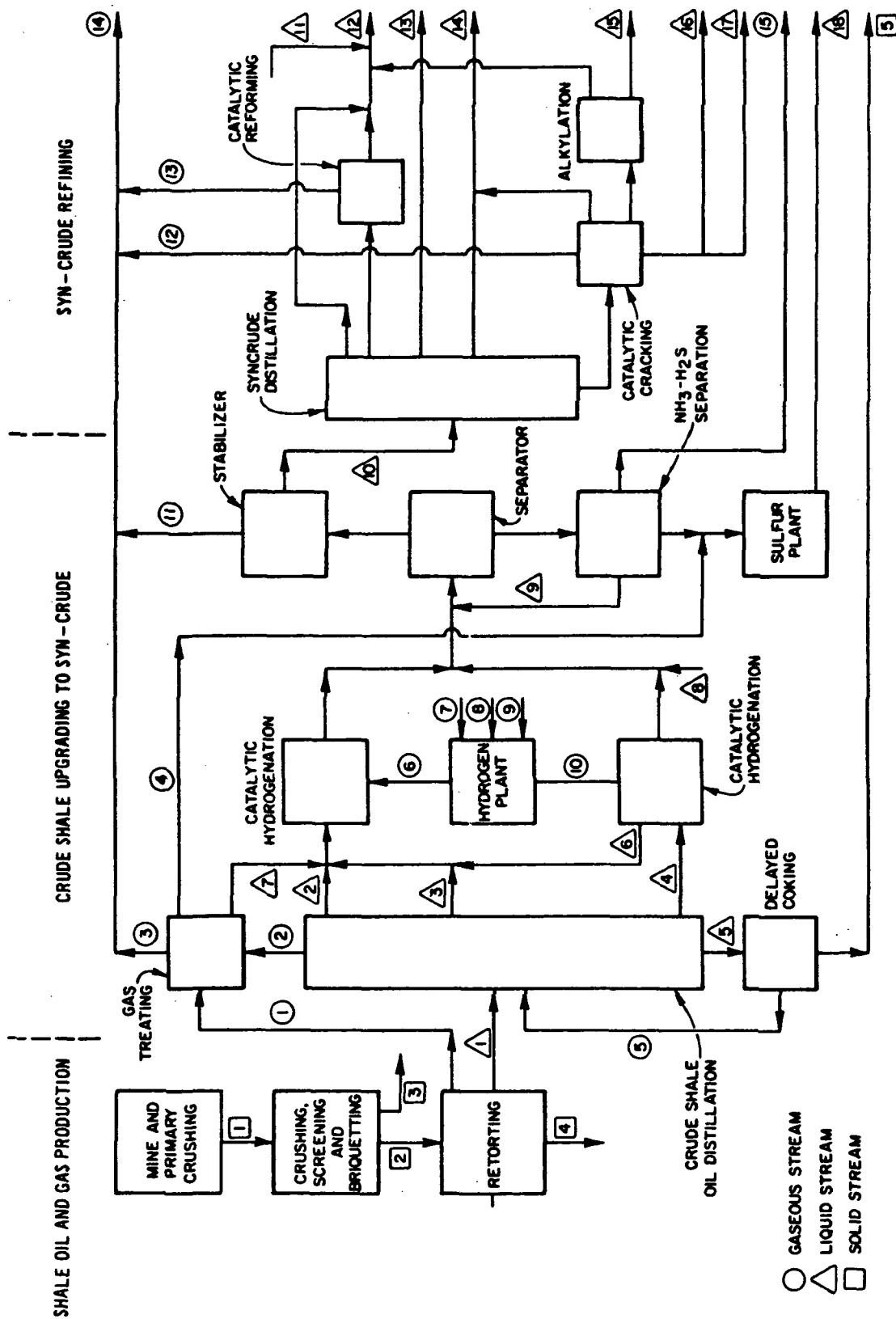


Figure 6-4. Flow Diagram for Production of Gasoline and Light Distillate (50,000 bbl/day) From Oil Shale (Reference 6-1)

Table 6-2. Process Energy Conversion Efficiencies for Various Fuels from Alternative Resources

Type of Plant	Energy Resource Used	Conversion Efficiency, Including All Products	Specific Conversion Efficiency			Source
			Gasoline	Distillate	Methanol	
Oil refinery	Crude oil	91-94%	82%	89%	-	Ref. 6-8, 6-9
Coal conversion and distillate refining	Coal	61% (including distillate refining)	42%	67%	-	Ref. 6-1
	Coal	41-47%	-	-	40-46%	Ref. 6-1
Oil shale conversion and syncrude refining	Oil shale	68%	60% ⁽¹⁾	-	-	Ref. 6-1

(1) Gasoline and light distillates combined.

FUEL CONVERSION EFFICIENCIES

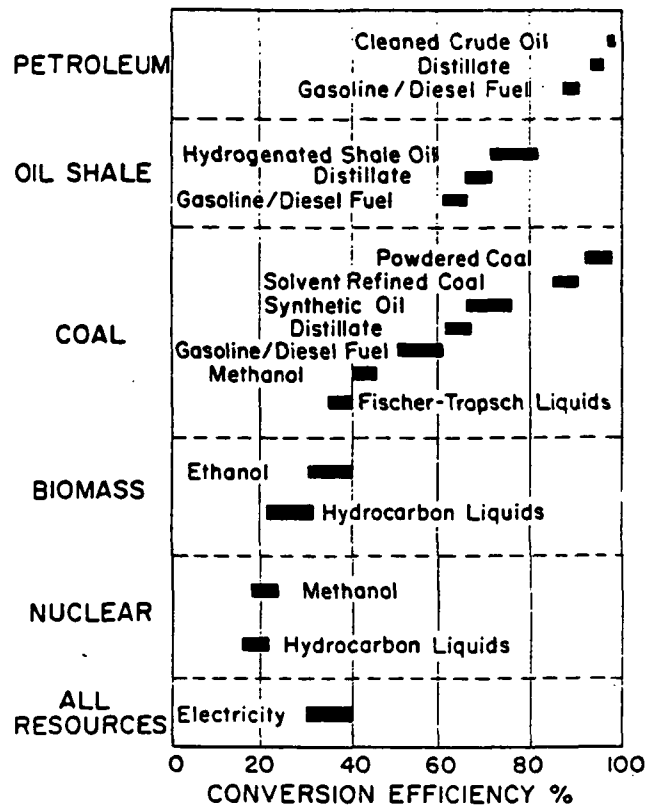


Figure 6-5. Processing Energy Required to Produce Different Fuels (Reference 6-8)

6.2 ENGINE FUEL REQUIREMENTS

6.2.1 Introduction

The major factors influencing the fuel requirements for an automotive engine from a combustion point-of-view are (1) method and character of ignition, (2) method of fuel metering and subsequent mixing with the air, (3) necessity for acceptable engine operation during warm-up, and (4) necessity for acceptable engine response during load transients. These factors can be translated into a set of properties or characteristics which a fuel must have in order that it can be used satisfactorily in a particular automotive engine. Fuels can be characterized in terms of many properties but the most important for the present purposes are (1) volatility, (2) octane number, and (3) cetane number. Volatility describes the ability of the fuel to change from the liquid to vapor state and depends on fractional composition, vapor pressure, surface tension, and heat of vaporization. Volatility is particularly important in carbureted and manifold or port injected engines. The octane number (RON or MON) is a measure of the tendency of the fuel to detonate after ignition. Detonation waves result from a

Table 6-3. Changes in Refinery Process Energy Use
for Various Product Fuel Mixes

Case	Gasoline	Diesel	Wide-cut	Process Energy-Use % of Crude Oil Input
Baseline	74	13.5	12.5	8.65
Unleaded gasoline	73	14.0	13.0	9.9
Max. Diesel	64	24	12.0	7.5
Max. Distillate	-	13	87.0	6.5

rapid buildup of flame speed and can be particularly troublesome in spark-ignited engines in which the fuel is well dispersed in the air before ignition. A high octane number indicates the fuel is resistant to detonation. The cetane number is a measure of the ability of the fuel to self-ignite with a minimum time delay when the ignition temperature of the fuel is exceeded. A high cetane number indicates that the fuel undergoes rapid self-ignition with only a short time delay. Engines which can operate satisfactorily in vehicles only when the volatility and octane or cetane number of the fuel are controlled within a narrow range are said to require high quality fuels. As noted in the previous section, such fuels involve complex refining operations and usually require significant process energy.

6.2.2 Conventional Engines

Conventional engines include spark-ignited, carbureted, or manifold/port injected reciprocating and rotary gasoline engines and naturally-aspirated and turbocharged diesel engines. The Honda CVCC stratified-charge engine is included in the spark-ignited, carbureted reciprocating gasoline group. Similarly, the Ford PROCO stratified-charge engine is included in the spark-ignited port fuel-injected group. It is included in this group even though, in the PROCO, fuel is injected when the compression stroke is well advanced whereas in the more conventional fuel-injected gasoline engines the fuel is injected primarily during the intake stroke. All of the conventional uniform-charge gasoline engines require a fuel with relatively high volatility and an octane number (RON) between 90-95. The stratified-charge gasoline engines require a fuel with a slightly lower octane number - possibly 85-90. Of the fuels being considered, the conventional gasoline engines

can burn gasoline (leaded or unleaded), alcohols (methanol and ethanol), and blends of gasoline and alcohol (References 6-11, 6-12). All of these are high quality fuels and require careful control in their production regardless of the input energy resource used.

Conventional diesel engines utilize compression-ignition, which occurs near the end of the compression stroke. Fuel is sprayed into the cylinder, and combustion must be initiated with a minimum of delay. This means that the cetane number of the fuel in a diesel engine should be controlled to a relatively narrow range (45-60) to achieve efficient combustion and to avoid smoking at high loads. Hence diesel fuel is also a high quality fuel in the sense that its properties must be carefully controlled. Methanol is not a suitable fuel for diesels and cannot be blended with diesel oil (Reference 6-13).

6.2.3 Advanced Engines

The advanced engines include the direct-injected, stratified-charge reciprocating and rotary engines, such as the Texaco TCCS and Curtiss-Wright rotary, the direct-injected, spark-assisted diesel engines, such as those being developed by MAN and Perkins (References 6-14, 6-15), gas turbine engines, and Stirling engines. In all of the advanced engines there is positive ignition by means of a spark-plug and/or pilot torch source, and the fuel is injected or transported to the flame zone almost immediately before burning. The result is that the advanced engines can burn almost any fuel and have no octane or cetane number requirements and only very limited requirements regarding fuel volatility. Hence the advanced engines can use the wide-cut distillates as well as gasoline, diesel oil, and alcohols. Except for the alcohols - which would require a fuel metering adjustment - the advanced engines can burn various fuels interchangeably and in blends with relatively small changes in thermal efficiency and emissions (References 6-16, 6-17).

Multifuel capability has been and continues to be of great importance in military applications, where high quality fuels are not always available. In the civilian sectors, except possibly in farming and construction, such fuels are readily available, and there has not been a significant advantage to engines which do not require high quality fuels. With the recent need to obtain automotive fuels from resources other than crude oil, the situation is changing and engines which can burn lower quality fuels are becoming increasingly attractive.

6.3 POSSIBLE ALTERNATIVE FUEL/ENGINE SCENARIOS

The previous consideration of engine fuel requirements and fuel production from alternative energy resources indicates that there will be significant differences in overall energy-use efficiency for various resource/fuel/engine combinations. That this is indeed the case is shown in Table 6-4, where the energy-use efficiencies for the various combinations are compared. It is clear from the table that the use of

Table 6-4. Comparisons of Energy-Use Efficiencies for Various Resource/Fuel/Engine Combinations

Resource	Fuel	Engine	Thermal Efficiencies, %		
			Resource Conversion	Engine	Overall
Crude oil	Gasoline	Uniform-charge Otto	90	16	14.4
	Diesel oil	Turbocharged diesel	92	20	18.4
	Distillates	Direct-injection stratified charge ⁽¹⁾	94	23	21.6
	Distillates	Gas turbine ⁽¹⁾	94	20	18.8
	Distillates	Stirling ⁽¹⁾	94	25	23.5
Shale oil	Gasoline	Stratified-charge Otto	65	18	11.7
	Diesel oil	Turbocharged diesel	67	22	14.7
	Distillates	Advanced engines ⁽¹⁾	70	25	17.5
Coal	Gasoline	Stratified-charge Otto	55	18	9.9
	Methanol	Stratified-charge Otto	45	18	8.1
	Distillates	Advanced engines ⁽¹⁾	67	25	16.8

⁽¹⁾ Advanced engines include direct-injection stratified-charge, gas turbine, and Stirling engines.

the advanced engines and wide-cut distillate fuels would result in significant energy savings. This is particularly true if the automotive fuels are produced from coal. Note also from Table 6-4 that converting coal to methanol and using it in a stratified-charge Otto engine is the least efficient option of all those considered.

SECTION 7

MATERIALS (METALS), MANUFACTURABILITY, AND COST

7.1 INTRODUCTION

Historically, automotive industry products have been manufactured in high volume and at a low unit cost. The production of automotive products requires a manufacturing system of nearly total automation coupled with computerized controls that maintain complete control of processing commencing with raw material and continuing through final assembly and test. A new component, to be considered as a candidate for introduction in the automotive industry, must exhibit two essential characteristics - namely:

- (1) Clear superiority in at least one area over the component it will replace (power to weight ratio, brake specific fuel consumption, packaging, etc.)
- (2) Manufacturing superiority with regard to ease of automation, labor content, and material cost yielding a total product cost that is competitive.

This section is concerned with the manufacturability of heat engines and their cost - which in the final analysis is a measure of their manufacturability.

Quoting from Reference 7-1:

The fundamentals of mass production have been defined as precision, standardization, interchangeability, synchronization, and continuity ... To achieve low cost at the end of a process, a heavy investment in plant, machinery, and tools must be made at the beginning. The technique cannot be economically employed under any and all conditions; the investment is justified only if a mass market exists. In other words, mass production and mass consumption are completely interdependent; ...

To reap the economic gains that may accrue from the mass production of any item, it is necessary that machinery investment costs be substituted for direct labor costs. Further, it is imperative that the manufacturing process be synchronized and continuous. Care must be exercised to insure that the sum total of all costs, which include both the unit variable cost (direct labor and materials) and the production facility and machine tool costs, are not greater than the comparable costs of the item that is being replaced. In the event that the item under consideration has a higher cost than the product being replaced, the net result could be either a higher selling price or a lower profit - or a combination of both.

The objective of this section is to assess the manufacturability and cost of producing alternative heat engines. In addition, it discusses the materials and processes that could be employed in producing them. The description of manufacturing methods is used to develop elemental cost estimates that are combined to form the necessary manufacturing costs and subsequently the selling prices for these engines.

Two methods for developing automotive engine manufacturing costs and one method for calculating selling price from manufacturing cost are presented. These methods are applied to obtain cost estimates for the Otto engine, the free-turbine Brayton engine, and the Stirling engine. The JPL descriptions of the configurations of these engines are discussed further in Section 7.2. All estimates are based on mass production volumes and techniques discussed previously.

This section draws upon the prior JPL engine costing effort as documented in Reference 7-2 and upon subsequent critiques from the automotive community (Reference 7-3). In addition, numerous fact-finding visits to both domestic and foreign automotive engine developers were conducted as part of this project.⁽¹⁾ Continued changes in engine design have occurred, and improved materials and updated process cost data have become available. All of these factors were taken into account in the development and application of this costing methodology.

The role of materials in improved heat engine applications is central. The potentially high thermal efficiency of Brayton and Stirling engines will only be realized with higher operating temperatures. The maximum allowable operating temperature for a given heat engine is dictated by the materials employed. A large portion of the engine cost is thus dependent upon materials required and the manufacturing techniques utilized to fabricate the component parts from those materials.

This section includes a general discussion of the highlights of recent developments in metallic materials technology and fabrication processes as they apply to the manufacture of Brayton and Stirling engines. Discussion of structural ceramic materials which have the capability of operating at temperatures hundreds of degrees higher than metallic alloys will be discussed in Section 8. Ceramic regenerators, which are required for the metallic Brayton and Stirling engines, are also assessed in the next section.

7.2 CONFIGURATIONS EVALUATED

The Brayton and Stirling engines evaluated in this section will be described briefly. The free-turbine or two-shaft Brayton configuration was selected for evaluation since it exhibits promise for the earliest introduction. It should be noted, however, that even this

⁽¹⁾ See Note, p. 7-39.

turbine configuration requires further development prior to being mass produced. The configuration of the free-turbine Brayton engine, as represented in Table 7-1, constitutes a projected production version, based on present day technology. The Stirling engine selected for evaluation is described in Table 7-2 and is based on present day technology.

Table 7-1. Salient Features of the Free-Turbine Brayton Engine

Maximum pressure ratio	4:1
Maximum turbine inlet temperature	1900°F
<u>Compressor</u>	
Type	Backswept centrifugal
Material	Aluminum
Variable inlet guide vanes	Yes
Design point efficiency	80%
<u>Gasifier Turbine</u>	
Type	Radial
Material	Superalloy
Design point efficiency	85%
<u>Power Turbine</u>	
Type	Axial
Material	Superalloy
Variable nozzles	Yes
Design point efficiency	85%
<u>Combustor</u>	
Type	Premix/prevaporizing (variable geometry)
Material	Steel and superalloy
<u>Regenerator</u>	
Material	Aluminum silicate (AS) or magnesium aluminum silicate (MAS)
Maximum inlet temperature	1800°F
Leakage	3%
Design point efficiency	90%
<u>Controls</u>	
Type	Integrated
Actuation	Hydromechanical
Logic	Analog

Table 7-2. Salient Features of the Stirling Engine

Working fluid	H ₂
Max. temperature	800°C
Max. pressure	200 atm.
Heater head	Minimum superalloy, tube bundle or investment cast construction
Cycle regenerator	Bonded metal fiber or metal foam or ceramic (MAS or AS)
Cycle cooler	Aluminum cross-flow heat exchanger
Pistons	Stainless steel
Rod seal	Sliding seal or roll-sock system
Block	Aluminum
Swashplate	Fixed; nodular iron
Preheater	Magnesium aluminum silicate (MAS) or aluminum silicate (AS)
Power control system	Mean-pressure-level regulation augmented with dead volume control

7.3 METAL TECHNOLOGY

Major programs dealing with metallic heat engines such as the DOE/Chrysler Gas Turbine Development Program and the DOE/Ford Stirling Engine Program are directed towards the development of engines which demonstrate improved fuel economy and reduced emission as compared to conventional Otto engines. These programs are dependent upon many technical disciplines, among which is metal technology.

7.3.1 Metallic Materials Technology

Nickel or cobalt based alloys which demonstrate high temperature strength capabilities, as well as suitable creep, low-cycle fatigue, and oxidation resistance at elevated temperatures are typically referred to as superalloys. These alloys can be categorized into groups of materials designated as Inconels, Hastelloys, Waspalloys, Renes, INs, and

others. Superalloys contain other alloying elements in addition to nickel and cobalt, such as chromium, molybdenum, tungsten, and others, depending on the specific alloy.

Superalloy technology has been heavily investigated and supported by large investments of research and development efforts since World War II. Hence, the technology is considered to be at a mature stage of development. Significant breakthroughs which would raise the service temperature of these materials more than a few degrees a year are not expected. Since cooling is not considered practical for passenger car application, the current operating temperature limit for uncooled superalloy turbine wheels and other stressed engine components is in the range of 1900°F with short transients in excess of 1900°F.

Protective coating systems are employed on the hot gas path components of aircraft gas turbine engines to permit operation with high transient temperatures. These coatings improve the thermal fatigue and hot corrosion resistance of the metallic substrate. The design limitations for rotating high temperature parts (i.e., a turbine wheel) are governed by the interaction of the creep and low cycle fatigue resistance properties of the alloy. Coatings do not improve these material properties. However, they do improve the corrosion resistance, which is beneficial.

Two critical problems arise when metallic alloys are subjected to high temperature, high pressure hydrogen environments as required for Stirling engine components. They are hydrogen embrittlement and hydrogen permeation. At ambient temperature, most high strength alloys, with good oxidation resistance, can exhibit hydrogen embrittlement under certain stress conditions (Reference 7-4), resulting in mechanical degradation such as loss in ductility, and reduction in strength and rupture life. At elevated temperatures some of these materials, such as austenitic alloys, have not shown susceptibility to hydrogen embrittlement. However, long term hydrogen embrittlement effects on specific superalloys require future study and quantification. The problem of hydrogen permeation through metals is critical since control of hydrogen loss is essential to the operational life between recharges of a Stirling engine. The permeability of superalloys being considered for Stirling engine heater head application is not well documented in published literature. A contract has been awarded to the Illinois Institute of Technology Research Institute to investigate and quantify the hydrogen permeation through selected alloys, and to evaluate the effectiveness of ceramic coatings such as silicon nitride.

7.4 MANUFACTURING OF HEAT ENGINES

Heat engine components are manufactured utilizing one or more of the following processes: casting, forging, stamping, fabrication, and purchased items. After receipt of the processed and purchased materials at an engine plant, the components are machined, as required, and then assembled into engines.

7.4.1 Conventional Engines

Automotive engines have a history of being producible at a low unit cost. The production of these engines requires the use of complex manufacturing facilities at the engine plant itself as well as by outside vendors at their plants. The following major operations are performed prior to arrival of parts and/or components at the engine manufacturing and assembly plant:

- (1) Foundry-manufacture of castings for engine cylinder blocks, cylinder heads, intake and exhaust manifolds, flywheels, pistons, piston rings, connecting rods, crankshaft, camshaft, etc.
- (2) Forge shop manufacture of valves, gears, drive parts, etc.

In addition, some parts are fabricated from several pieces or are purchased items —

- (3) Valve rocker covers, oil pan, water pump, fuel pump, radiator, fuel control system, alternator, voltage regulator, starting motor, coil, distributor, spark plugs, wires, muffler, catalytic converter, exhaust gas recirculation system, bearings, valves, valve lifters, springs, etc.

When the preprocessed and purchased components are received at the engine plant, they are machined, as required, and then used to assemble complete engines. The following description of the machining and assembly processes for cylinder blocks and crank shafts is presented as an example of engine manufacturing operations.

Locating surfaces are machined on the cylinder blocks and then the cylinder blocks are conveyed to a broaching machine. The flat planar surfaces, which mate with other major engine components, are machined. The blocks are then transferred to a machine which bores the cylinders. The next operation is to drill and tap holes for attaching mating parts to the block. The block is then transferred to a honing machine where the cylinders are honed to the final finish and dimension. At this point the cylinders can be inspected for bore measurement. This bore measurement information results in the marking of each cylinder and the future use of graded pistons and rings for a selective fit with each cylinder bore. The cylinder blocks are now conveyed to a line, where assembly of the complete engine takes place.

Crankshaft machining is precise and critical to a smooth and quiet performing engine. The cast part is machined and then ground on a series of centers to provide smooth and concentric surfaces for the main bearings and also for the connecting rod bearings.

In general, the manufacturing process for conventional engines requires that a large number of surfaces be machined to relatively close tolerances and that relatively large amounts of metal be removed from castings, forgings, etc. Since each engine has a multiplicity of cylinders, many operations are identical and therefore are executed simultaneously on multiple spindle machines. Many of the components for the engine such as pistons, piston rings, connecting rods, valves, valve lifters, rock arms, springs, etc., are identical and are produced at high speeds with repetitive operations. The majority of components are made from gray cast iron or low alloy steel and are machined at high speed, without regard to excessive tool wear. The design of most parts incorporates cylindrical or flat surfaces that can be machined with a high degree of accuracy without requiring complex machining operations. This is the result of part designs that are purposely kept simple.

Quality assurance is effected by inspecting the equipment and tools on a regular basis. When the tooling has been certified as being correct, a few pieces are produced and then inspected to verify that the drawing requirements have been met. Once these parts have been approved as being acceptable, the automated production of parts can begin. Periodic inspection of parts and tooling insures that quality control standards are being met.

7.4.2 Brayton Engines

Brayton engines designed for use in passenger automobiles are composed of the following basic components:

- Compressor assembly
- Turbine assembly (ies)
- Regenerator assembly (ies)
- Combustor assembly
- Control system

Table 7-3 is a parts breakdown for metal free-turbine Brayton engines of 100 hp and 150 hp. An estimate of the weight along with a forecast of the material and manufacturing process requirements is depicted. The material for each critical component is listed by material family (Table 7-4), and a process coding tabulation (Table 7-5) is employed to denote the required manufacturing process.

The component and engine weights, shown herein, are finished weights, without an allowance for scrap. Brayton engines require the use of high temperature materials such as stainless steels and super-alloys for the turbine and combustor assemblies as well as ceramics for

Table 7-3. Free-Turbine Brayton Engine Parts Breakdown

Component/Subassembly	100 HP			150 HP		
	Weight, lb	Mat'l (1)	Proc. (2)	Weight, lb	Mat'l (1)	Proc. (2)
<u>Power Turbine Assembly</u>	(95.1)			(110.8)		
Turbine wheel	2.4	H	62	3.0	H	62
Shaft	1.0	E	69	1.0	E	69
Inlet housing	5.0	H	62	5.4	H	62
Discharge shroud	5.0	E	34	6.0	E	34
Outer housing	50.0	A	61	62.0	A	61
Bearings (3)	0.5	D	07	0.5	D	07
Variable nozzles	2.0	H	62	2.0	H	62
	1.2	G	69	1.2	G	69
	2.5	F	08	2.5	F	08
	3.5	E	69	3.5	E	69
Actuator	2.0	Z	07	3.0	Z	07
Lubricant, pump, tank	6.0	Z	07	6.7	Z	07
Fasteners, seals, etc.	12.0	Z	07	12.0	Z	07
	2.0	G	07	2.0	G	07
<u>Gasifier Turbine Assembly</u>	(26.0)			(27.4)		
Turbine wheel	4.0	H	62	4.7	H	62
Shaft	1.0	G	08	1.2	G	08
Inlet scroll	5.5	E	61	6.0	E	61

Table 7-3. Free-Turbine Brayton Engine Parts Breakdown (Continuation 1)

Component/Subassembly	100 HP				150 HP			
	Weight, lb	Mat'l (1)	Proc. (2)		Weight, lb	Mat'l (1)	Proc. (2)	
Discharge shroud	2.5	E	62		2.5	E	62	
Outer housing	Included in Power Turbine Assembly				Included in Power Turbine Assembly			
Bearing	Included in Compressor Assy.				Included in Compressor Assy.			
Fasteners, seals, etc.	12	Z	07		12	Z	07	
	1	G	07		1	G	07	
<u>Combustor Assembly</u>	(34.8)				(39.0)			
Liner	1.0	H	54		1.0	H	54	
Atomizer/vaporizer dome Assembly	2.0	E	24		2.2	E	24	
Secondary air control Housing	0.5	E	74		0.5	E	74	
Sensors and actuators	26.0	A	61		30.0	A	61	
Ignition assembly	2.1	Z	00		2.1	Z	00	
Seals, fasteners, etc.	2.2	Z	07		2.2	Z	07	
	1.0	G	07		1.0	G	07	
<u>Regenerator Assembly</u>	(60.0)				(66.0)			
Core	19	I	76		20	I	76	
Housing and ducting	28	Z	61		33	Z	61	
Seals	3	Z	07		3	Z	07	
Gear	5	D	61		5	D	61	
Motors, fasteners	5	Z	07		5	Z	07	

Table 7-3. Free-Turbine Brayton Engine Breakdown (Continuation 2)

Component/Subassembly	100 HP			150 HP		
	Weight, lb	Mat'l (1)	Proc. (2)	Weight, lb	Mat'l (1)	Proc. (2)
<u>Compressor Assembly</u>	(55.6)			(63.6)		
Impeller	2.0	J	62	2.0	J	62
Shaft	0.6	G	69	0.6	G	69
Diffuser	12.5	A	61	15.0	A	61
VIGV assembly	6.5	Z	00	7.6	Z	00
Actuator	1.0	Z	00	1.2	Z	00
Housing	21.0	A	61	24.0	A	61
Air bearings	2.0	D	07	2.2	D	07
Seals and fasteners	10.0	Z	07	11.0	Z	07
<u>Advisory Drive</u>	(19.0)	Z	80	(19.0)	Z	80
<u>Control System</u>	(39.0)			(39.0)		
Electronics and sensors	---			---		
Fuel valves, pump filter	21.0	Z	07	21.0	Z	07
Misc. components	18.0	Z	07	18.0	Z	07
<u>Air Inlet Assembly</u>	(12.5)			(14.0)		
Housing and ducting	8.5	L	07	10.0	L	07
Air cleaner	4.0	Z	07	4.0	Z	07

Table 7-3. Free-Turbine Brayton Engine Breakdown (Continuation 3)

Component Subassembly	100 HP			150 HP		
	Weight, lb	Mat'l (1)	Proc. (2)	Weight, lb	Mat'l (1)	Proc. (2)
<u>Reduction Drive Assembly</u>	(49.0)			(57.0)		
Gears, shafts, bearings	13.0	D	80	15.0	D	80
Housing	36.0	A	61	42.0	A	61
<u>Auxiliaries</u>	(37.0)			(37.0)		
Starter	20.0	Z	07	20.0	Z	07
Alternator	17.0	Z	07	17.0	Z	07
Total, Engine Ready-to-Run	(428.0)			(472.8)		
<u>Battery</u>	(42.0)	Z	07	(42.0)	Z	07
<u>Transmission</u>	(130)	A,J	80	(130.0)	A,J	80
(Conventional 3-speed automatic)						
Total, Power System	(600.0)			(644.8)		
(1) Material type code (from Table 7-4)						
(2) Process code (from Table 7-5)						

Table 7-4. Material Type Codes

A.	Cast iron (all types, gray, malleable, nodular, ductile, GM-60, NIRST, etc.)
B.	Carbon steel (AISI grades, etc.)
C.	High carbon steel (AISI grades, etc.)
D.	Alloy steel (AISI grades, includes carburizing and nitriding grades, etc.)
E.	Austenitic stainless steel (300 series, 200 series, special modifications such as CRM-6D, etc.)
F.	Ferritic or martensitic stainless steel (400 series, etc.)
G.	Precipitation hardening stainless steel (A-286, 17-4 PH, 15-5 PH, etc.)
H.	High temperature superalloys (Inconels, Hastelloys, Waspalloys, Renes, INs, 713LC, Multimet, other specialty alloys, etc.)
I.	Ceramic
J.	Aluminum alloy
K.	Copper alloy
L.	Plastic or Teflon or Rulon
M.	Rubber or elastomeric material
Z	Not broken down (conventional auto materials) ⁽¹⁾ nonhomogeneous or miscellaneous and purchased parts

(1) Conventional auto materials do not include E, F, G, H, and I, but do include additional quantities of some of the others explicitly listed.

the regenerators. The balance of the engine components can be fabricated with conventional automotive materials and processes. The critical high temperature components and their manufacturability are discussed in the following sections.

Table 7-5. Process Codes

10	Brazed	01	Casting
20	Welded	02	Investment casting
30	Formed	03	Forging
40	Unassigned code	04	Sheet
50	Stamped	05	Plate
60	Machined	06	Ceramic concrecence ⁽¹⁾
70	Fabricated	07	Purchased
80	Mechanically assembled	08	Tubing
00	Nonhomogenous assembly or miscellaneous	09	Bar stock

-
- (1) The term "concrecence", as used in this study, is a generic one referring to any existing or potential fabrication process which yields a completed ceramic shape or integral structure.
-

7.4.2.1 Turbine Wheels. The present state of the art can produce an uncooled, uncoated superalloy turbine wheel of passenger car size by the investment casting process.

The investment or lost wax coating process is currently being used to manufacture turbocharger components. The process is complicated, requires a large amount of labor, and utilizes expensive materials that at the present time are generally not recoverable. While this process has been found to be cost-effective for aircraft type engine parts and turbosuperchargers, it is considered to be one of the most critical cost factors in developing gas turbine engines for passenger automobiles. However, the process has much to offer the gas turbine industry, e.g., excellent surface finish, dimensional control of sophisticated shapes, and in some cases, the only method of casting some configurations.

For large volume production of investment cast parts, a metal die is made from which a wax pattern is produced by the injection molding process. The wax patterns are then manually dipped into successively coarser ceramic slurries to form a surrounding ceramic mold. An average of seven dips and drying cycles is required to form the ceramic mold. The mold is then cured in an oven and the wax is melted out at the same time. Next, the mold is preheated to within 300°F of the melting temperature of the metal to be poured. The mold is then brought to a vacuum furnace operating at approximately 2 microns Hg pressure, and the molten metal is poured into the mold. After casting, the ceramic mold is broken off the casting periphery by a pneumatic hammer. The casting is then manually sandblasted to remove any remaining ceramic and is ready for inspection. Inspection involves dipping into a fluorescent dye penetrant (zyglo) to discover surface flaws, visual inspection by gauging for specified profiles, and X-raying for internal flaws.

New and faster drying precoats coupled with faster drying ceramic slurries and completely automatic handling equipment must be developed and/or improved. The recovery and reprocessing of refractories may offer possibilities for cost reductions.

An improvement to the traditional investment casting process is the AIREFRACTM process developed by the Air Research Casting Company, a division of the Garrett Corporation. This process employs a reusable rubber pattern coupled with the reuse of some previously expendable foundry materials. The AIREFRACTM technique appears to be more amenable to mass production, is faster, requires fewer labor hours, and is lower in cost.(1)

A current program sponsored by DOE and conducted by the Government Products Division of Pratt and Whitney Aircraft is aimed at demonstrating manufacturing techniques utilizing the GATORIZINGTM forging process to produce superalloy gas turbine rotors which are both aerodynamically efficient and economically feasible for large volume production (References 7-6, 7-7). The GATORIZINGTM process is an isothermal, super-plastic forging process whereby a superalloy billet is plastically deformed under pressure and temperature into a given configuration.

The fabrication of a turbine rotor utilizing this process requires two forging steps. The first step produces a nonbladed disk preform from a cylindrical billet. The preform is then forged to a finish dimension disk and the blades extruded onto the rim of the disk to form the integral bladed rotor. The rotor is then heat treated to optimize the mechanical properties followed by final machining to debur the forging and grind the blades to the correct length.

(1) Source: Reference 7-5, and a fact-finding visit by ATSP team members to Garrett Corporation, July 1977.

The turbine blade configuration is the limiting factor in this forging process. The geometric restrictions are an interrelated function of blade length, thickness, and radial taper. Care must be exercised to insure complete fill of the blade die cavities (References 7-6, 7-7).

The cost of finished superalloy parts can be reduced by increased recycling of superalloys. Currently, most specifications require superalloy parts to be made only from virgin materials. This requirement may work for low volume, high reliability spacecraft applications, etc., but is not economical for automotive industry products. When superalloys are used in automotive applications, it will be expected that the specifications will permit and encourage recycling of superalloy scrap in order to reduce both costs and mining requirements of superalloy constituents.

7.4.2.2 Stainless Steel Parts. The inlet scroll and discharge shroud can be fabricated from stainless steel. The following units can also be fabricated from stainless steel:

Atomizer/vaporizing dome assembly

Secondary-air control components

Variable inlet guide-vane assembly

7.4.2.3 Combustor Liner. This component is fabricated from superalloys.

7.4.2.4 Compressor Impeller. An aluminum material is used for this part which is made by the investment casting process described previously.

7.4.2.5 Regenerators. Since the regenerators are fabricated ceramic parts, they are discussed in Section 8.

7.4.2.6 Fuel Control Units. The technology of fuel control units for automotive gas turbines is still in a state of change and improvement, but no unusual mass production problem is anticipated for these units.

7.4.2.7 Components Produced by Using Conventional Materials and Processes. The remaining parts and subassemblies required for the free-turbine Brayton engines, which consist of the accessory drive, air inlet assembly, speed reducer assembly, and auxiliaries are all of conventional automotive materials and processes. None of these items requires any advanced materials or processes for their production.

7.4.3 Stirling Engines

The Stirling engine as presently configured for use in passenger automobiles is composed of the following basic items:

Combustor	Linkage mechanism and bearings
Heater head	Seals
Block and pistons	Control systems - air/fuel and power
Preheater	Cooling system
Cycle regenerators	
Cycle coolers	

Table 7-6 lists the Stirling engine components for both the 100 hp and 150 hp metallic configurations. Table 7-6 contains a weight estimate as well as a forecast of material and manufacturing process requirements. The material for each major component is listed by material family (Table 7-4) and a process coding tabulation (Table 7-5) is employed to denote the required manufacturing process. The component and engine weights shown herein are finished part weights, without an allowance for scrap.

Since the Stirling engine design and development for use as an automotive engine is in its early stage, the selection of materials and/or processes required to mass produce the various components is still under development. Consequently, the following discussion will concentrate on forecasting the materials and processes that may be employed for the various components.

7.4.3.1 Heater Head Assembly. This assembly consists of cylinders joined to regenerator housings by means of thin-walled tubes that are brazed to each unit. The heater head assembly is the most difficult item to produce for this engine. Hot combustion gases flow past the tubes and transfer their heat to the pressurized, hermetically sealed hydrogen gas. The material chosen for these heater tubes must exhibit good high temperature creep and stress-rupture properties, and high resistance to oxidation, and be able to resist hydrogen embrittlement. At the present time, Multimet N-155 is used in the fabrication of these tubes. It was selected because it meets the technical requirements, and for a superalloy is relatively low in cost. Since N-155 has a high percentage of iron, it can be made at a lower cost than the Inconels or Hastelloys, which require greater amounts of expensive elements such as nickel, chromium, and cobalt.

Table 7-6. Stirling Engine-Metallic Configuration

Component/Subassembly	150/170 HP	100 HP	Material ⁽¹⁾	Process ⁽²⁾
	4-215 Weight, lb	4-98 Weight, lb		
<u>Heater Head Subassy.</u>	123	98	--	12
Cylinder (4)	20	16	E	63
Regen. hsg. (8)	25	20	E	63
Heater tubes	14	11	H	08
Fins	9	7	E	54
Bottom plate & ring	30	24	B	65
Heater hsg.	25	20	B	63
<u>Combustor Subassy.</u>	5.0	4.0	--	80
Burner can	1.0	0.8	H	64
Atomizer assy.	4.0	3.2	Z	80
<u>Top Cover Insulation, Attachment & Misc.</u>	20	16	B	54
<u>Internal Cycle Regen. Cooler Assy (8)</u>	41	32		
Regenerator (8)	10	18	E	07
Cooler (8)	15	12	J	18
Attachment, misc.	16	12	B	07
<u>Piston Assy (4)</u>	26	21.5		
Dome (4)	9	7.5	E	02
Ring (4)	--	--	L	07
Foot (4)	11	9	E	22
Attach & misc.	6	5	B	07
<u>Rod Seal Assy.</u>	11	11		
Guides (4)	10	10	J	69
Roll-sock (4)	0.08	0.08	M	07
Pump rings (4)	0.08	0.08	Z	07
Reg. valve (4)	0.8	0.8	B	07
Sliding seal (8)	--	--	M	07
<u>Swashplate Drive Assy.</u>	31	24		
Crossheads (4)	22	17	A	61
Sliders	9	7	D	69

Table 7-6. Stirling Engine-Metallic Configuration (Continuation 1)

Component/Subassembly	150/170 HP	100 HP	Material ⁽¹⁾	Process ⁽¹⁾
	4-215 Weight, lb	4-98 Weight, lb		
<u>Mainshaft Assy</u>	69	54		
Shaft	28	22	A	61
Swashplate	34	27	A	61
Main bearing	7	5	D	07
<u>Cylinder Block Assy</u>	115	90		
Crosshead block	32	24	J	61
High press. block	65	50	J	61
Cylinder liners	2	2	D	68
Oil pump	4	4	Z	07
Attach & misc.	6	5	B	07
Oil	6	5	Z	07
<u>Combustor Auxiliaries</u>	41	35.5		
Igniter plug & safety valve	2	2	Z	07
Blower ducts	4	3.5	B	07
Blower & pulley	11	9	Z	07
Blower motor	18	15	Z	07
Air silencer & filter	3	3	Z	07
Misc.	3	3	Z	07
<u>Preheater Assy</u>	83	71		
Core	26	21	I	76
Gears & drive	17	14	Z	07
Bearing	2	2		
Cover	22	18	B	54
Seals	13	13	Z	07
Pulley	3	3	A	61
<u>Coolant Pump & Drive</u>	22	18	J	61
<u>Power Control System</u>	40	40		
Pressure control	40	40	Z	07
<u>Fuel Supply Controls</u>	9	9	Z	07
<u>Accessory Drive System</u>	35	32	Z	07

Table 7-6. Stirling Engine-Metallic Configuration (Continuation 2)

Component/Subassembly	150/170 HP	100 HP	(1)	(2)
	4-215 Weight, lb	4-98 Weight, lb		
<u>Engine Auxiliaries</u>	52	52		
Starter	20	20	Z	07
Alternator & voltage reg.	17	17	Z	07
Misc.	15	15	Z	07
<u>Radiator Assy</u>	122	98		
Radiator	55	44	J	18
Fan & shroud	15	12	B	54
Coolant	52	42	B	07
<u>Total - Engine Ready to Run</u>	845	706		
<u>Battery</u>	45	40	Z	07
<u>Transmission - 3-Speed Automatic</u>	175	150	B-J	80
<u>Total - Power System</u>	1065	896		

(1) Material Type Code from Table 7-4.

(2) Process code from Table 7-5.

The N-155 heater head tubes are hand assembled to the mating part and brazed together with a microbrazing type alloy. This process is costly, and time consuming, but readily adaptable to volume production. The design and assembly of heater head tubes to the mating part requires investigation of many welding and brazing techniques, including electron beam, laser, etc.

The present state-of-the-art requires that fins be attached to the thin-walled tubes to provide for heat transfer to the sealed hydrogen. A Type 310 austenitic stainless steel can be used for these fins and parts can be stamped from sheet stock and then brazed to the tubes.

The heater head must be designed so that excessive leakage or diffusion loss of the pressurized hydrogen gas is prevented. The heater head tube walls provide the greatest avenue for hydrogen diffusion losses. Since the cylinder walls are thicker and cooler than the tube walls, hydrogen diffusion through these parts is not a significant problem. If hydrogen diffusion develops into a major problem, the use of a barrier coating, such as silicon nitride, could be employed.

Extensive design and manufacturing studies need to be undertaken to determine the optimum configuration for the heater head assembly. Such studies should include the feasibility of making an integral casting by sand, permanent mold, investment or other methods. Future developments in ceramic technology could result in the Stirling engine heater head configured as a monolithic ceramic component. The H₂-permeation resistance of ceramics is known to be superior to that of the superalloys, but it has not been demonstrated that even silicon nitride can adequately contain H₂ at 2500°F material temperature.

Various combinations of cylinders and regenerator housings are possible:

- (1) Four cylinders per casting, plus four regenerator housings per casting.
- (2) Cylinder-regenerator housing pairs.
- (3) Heater head quadrants.

Various material compositions require additional study for castability, weldability, and cost.

The cylinders and regenerator housings for prototype engines are investment cast from Haynes 25 superalloy. Production engines could use CRM-6D, which is a modified austenitic 300 series stainless steel, for the cylinders and regenerator housings. The feasibility of fabricating a composite cylinder and regenerator housing should be determined. The possibility of welding or brazing parts of cast, forged, or drawn cylinders with regenerator housings of dissimilar materials requires assessment. The upper hot part of the cylinder might be made of high temperature nickel base alloy while the lower cooler parts could be made from lower cost 300 or 400 series stainless steel.

7.4.3.2 Combustion Chamber. The combustion chamber for the Stirling engines is a burner-can that mounts inside the heater tube cage. This can is fabricated from superalloy sheet metal. Since the burner can is essentially unstressed, future configurations could be made from ceramic materials.

7.4.3.3 Cycle Regenerator. In order to attain a high efficiency regenerator, the current configurations use fine stainless steel (type 301) woven screens. This sintered gauze is expensive to manufacture.

Tests by Philips and United Stirling indicate that woven wire is superior to sintered powders, chopped wire, fine machined fibers, etc., because of its regularity of cross section, good control of the filling factor, and the ability to be fabricated with uniform density. Alternatively, the regenerators could be fabricated from thin alloy sheets having a regular pattern of holes for retention of screen properties. These could be produced by alloy electroforming or chemical milling of a rolled sheet after the application of a photoresist material. The cycle regenerator is an area for future design and manufacturing technique improvement.

7.4.3.4 Cycle Cooler. The Stirling engine cycle coolers are currently constructed of several hundred fine, closely spaced, drawn, stainless steel tubes, which are vacuum brazed into stainless steel headers. Future designs could substitute corrugated plate assemblies for the fine tubing and different fabrication techniques - such as flux brazing aluminum or copper alloy tubes to headers.

7.4.3.5 Piston Assembly. Pistons currently used in Stirling engines have a 300 series stainless steel dome that is electron beam welded to a stainless steel foot. Pistons could possibly be fabricated from ceramics in the future.

7.4.3.6 Cylinder Block Assembly. The cylinder block assembly consists of a crosshead block and the high pressure block (crankcase). The high pressure crankcase functions as a major structural element and a housing for the high pressure hydrogen distribution system. Currently, it is made by machining the main body from heavy blocks of 300 series stainless steel, attaching other parts by brazing with gold brazing alloy and then final machining. Future designs could utilize the following configurations:

- (1) An aluminum high pressure crankcase using alloy steel cylinder liners.
- (2) Forged steel components assembled with conventional auto industry welding and/or brazing techniques.

7.4.3.7 Preheater. The preheater currently in use on the Stirling engine is a rotating ceramic disc. It is similar to the regenerator core (MAS or AS) used in gas turbine engines. A discussion of ceramic materials appears in Section 8.

7.4.3.8 Swashplate Drive Assembly and Main Shaft Assembly. These components can all be mass produced by using conventional automobile industry processes. The current crosshead configurations are made from alloy steel, but could be fabricated from cast iron. The shaft could be made from modular cast iron.

7.4.3.9 Piston Rod Seal. The piston rod seal is one of the critical components in the Stirling engine. Currently, the Ford/Philips engines use a so-called "roll-sock" seal which is a thin, flexible (elastomeric), oil-supported, cylindrical (slightly conical) tube, which is rigidly affixed to both the piston rod and the adjacent wall, providing a hermetic seal. Despite the fact that this type of seal has proved to be effective as a hydrogen gas barrier, it requires a very serviceable material (polyurethane rubber) and a sensitive control system for maintaining the proper oil pressure. If the roll-sock seal fails, an extensive disassembly of the engine for its replacement is required.

An attractive alternative to the "roll-sock" seal is the sliding seal being developed by United Stirling. This seal is more suitable for high volume production and assembly. This arrangement has unsupported fluorocarbon rings placed in the piston rod guides that then bear upon a microfinished rod.

7.5 COSTING ASSUMPTIONS

Engine cost estimates are necessarily based on a number of assumptions regarding industry business practices. The assumptions made in the present work are listed below:

- (1) The engine plant is located in the U.S. with an annual production volume of 400,000 engines.
- (2) Tooling and equipment are investments with an economic lifetime of 10 years (the design life of a tool, not necessarily its use).
- (3) Facilities including buildings, plants, and related structures have an economic life of 20 years.
- (4) An acceptable rate of return on investment in tooling, equipment, and facilities for this industry is 15 percent before tax considerations.

The locations and production volume assumed agree with those of other reports by JPL (Reference 7-2) and Clayton (Reference 7-8, p.89) and with existing facilities. The lifetime assumed for tooling and equipment is consistent with the engine design cycle and accepted tool and equipment life (Reference 7-2). The facility lifetime of 20 years is in agreement with current industry practice. The assumption of a 15 percent rate of return on investment is arbitrary but reasonable for the automobile industry.

The costing methodology and application which follow must be viewed in the light of the foregoing assumptions. This is especially important when comparing the cost estimates presented here with other work.

Two methods were used to prepare estimates of the variable costs of engine production. The variable costs include material and direct labor costs. The first method involves costing the major subassemblies of each engine by relating each subassembly to presently produced or similar items. The subassembly variable costs are then summed to obtain the variable cost of the overall engine assembly. An error bound for the engine variable cost is determined by taking the square root of the sum of the squared subassembly error bounds. For parts without a conventional analog, the second method was used. In the second approach, materials used in engine manufacture are separated by type and weight. Variable-materials cost is then calculated by multiplying material weight times costs for each material required and summing up over all the materials involved in producing the engine subassembly. Major materials types and estimates of current costs are contained in Table 7-7. Labor cost including foundry labor, machining labor, and assembly labor is then added using a current average United Automobile Worker's labor rate of \$8.00 per hour⁽¹⁾ to determine the variable engine cost.

Engine variable production costs by major assemblies are given in Tables 7-8, 7-9, 7-10, and 7-11. These variable costs include direct labor and direct materials costs including purchased items. Indirect costs and overhead allocations are not included.

Table 7-8 contains variable production costs by major assembly for the carbureted gasoline engine with an oxidation catalyst. Except for an increase in cooling system costs, estimates were based on major assembly costs presented in Reference 7-2 adjusted for inflation (a 20 percent increase). The variable costs for the smaller four-cylinder engine were estimated from those of the larger six-cylinder engine. The larger engine was presumed to take an additional hour of machining labor (see Table 7-12).

Variable production costs for major assemblies for a fuel-injected gasoline engine using a three-way catalyst are given in Table 7-9. With respect to the costs presented in Reference 7-2, the fuel-injected engine had cooling system costs increased as well as the 20 percent adjustment for inflation. The 100 hp engine costs represent a reevaluation of the 85 hp engine of Reference 7-2.

Major assembly variable production costs for 100 and 150 hp free-turbine Brayton and Stirling engines are given in Tables 7-10 and 7-11. Ceramics are used only for insulation and for the regenerator core. The cost of the 150 hp Stirling engine was derived from the 170 bhp engine

⁽¹⁾ Source: Telecon with United Automobile Worker's Union, Los Angeles, CA, Sept. 1977.

Table 7-7. Automotive Engine Materials Costs⁽¹⁾ in 1977 Dollars

Material	Cost Range in \$/lb
Cast iron	0.15 - 0.20
Carbon steel	0.13 - 0.21
Alloy steel	0.15 - 0.25
Austenitic stainless steel	0.40 - 1.12
Ferritic stainless steel	0.31 - 0.68
Precipitation hardening steel	0.71 - 1.60
Superalloy ⁽²⁾	3.00 - 4.50
Ceramic ⁽³⁾	0.75 - 1.25
Aluminum alloy	0.60 - 0.65
Copper alloy	0.60 - 0.95

- (1) These are cost estimates for materials in ingot form, not including fabrication costs. They are taken from issues of American Metalworking News and Iron Age, July and August, 1977.
- (2) Cost projections for materials in mass production volumes taken from Reference 7-2.
- (3) Cost based on potential production in mass production quantities.

of Reference 7-2 and adjusted by 20 percent for inflation to 1977 dollars. The 150 hp engine was then scaled to 100 hp and the major assembly costs adjusted accordingly.

7.8 VARIABLE COSTS BY MATERIAL TYPE AND LABOR

Variable production costs derived from material types and labor expended are given for the four engine types in Table 7-13. The breakdown of direct labor by category is given in Table 7-12.

Table 7-8. Variable Costs for a Carbureted Gasoline Engine With Oxidation Catalyst

Major Component	Cost in 1977\$	
	100 HP (4 cyl.)	150 HP (6 cyl.)
Basic engine (block, heads, valves)	145 ± 10	190 ± 10
Ignition system	10	12
Induction and fuel system	40 ± 10	50 ± 10
Emission control	95 ± 25	95 ± 25
Cooling system	45	50
Auxiliaries	30	35
Total	365 ± 20	432 ± 30

The weights for the different material types used in the free-turbine Brayton and Stirling engines were derived from the detailed component weights given in Tables 7-3 and 7-6. The 150 hp Otto engine weights by material type were taken from Reference 7-2, with the 100 hp Otto engine material weights inferred accordingly. The material weights of Table 7-12 were multiplied by average material prices of Table 7-7 to obtain the material costs given in Table 7-12. The total engine variable costs calculated were within 5 percent of the engine variable costs calculated by subassembly.

Given the good agreement of engine variable costs determined by the two methods, it was decided to carry forward only one set of engine variable costs to determine engine selling prices. Since the engine variable costs based on material and labor types were based on more detail, the costs from Table 7-12 are used in all subsequent calculations.

7.9 TOOL, EQUIPMENT, AND FACILITIES COSTS

Up to this point, the only costs that have been presented have been the variable costs of materials and labor. There are several other costs of production that must be considered when computing the wholesale

Table 7-9. Variable Costs for a Fuel-Injected Gasoline Engine With 3-Way Catalyst

Major Component	Cost in 1977\$	
	100 HP (4 cyl.)	150 HP (6 cyl.)
Basic engine (block, heads, valves)	145 ± 10	190 ± 10
Induction system (manifold, injection pump, injectors, air cleaner, fuel pump, and filter)	95 ± 20	115 ± 40
Ignition system (electronic distributor, coils, and spark plugs)	10	12
Emission control system	105 ± 10	105 ± 10
Auxiliaries	30	35
Cooling system	45	50
Total	430 ± 25	507 ± 43

(OEM) price of automotive engines. Some of these costs stem from the facilities required to produce the engines.

The facilities needed to produce 400,000 engines per year include (1) direct production machinery consisting of transfer lines, standard and special machines, and material handling equipment; (2) buildings that house the production and foundry equipment; (3) foundry and forge capacity in three subcategories - (a) cast iron and steel, (b) superalloys and stainless steel, and (c) aluminum die casting; and (4) tooling (both at the engine plant and by various vendors). Table 7-14 lists the facilities required in the various categories for the alternative engines.

The data for Table 7-14 were primarily obtained from Reference 7-2, adjusted for inflation between November 1974 and July 1977. The recent announcement by Ford Motor Company (Reference 7-9) of plans to build a new engine plant in England for \$315 million provides a point of reference for the total engine costs for the gasoline Otto engine.

Table 7-10. Variable Costs for a Metallic Free-Turbine
Brayton Engine

Major Component	100 HP	150 HP
Power assembly	\$155 ± 45	\$180 ± 50
Gassifier turbine assembly	115 ± 35	135 ± 40
Combustor assembly	50 ± 15	60 ± 20
Regenerator assembly	110 ± 30	120 ± 30
Compressor assembly	45 ± 25	55 ± 30
Accessory drive	20 ± 10	25 ± 10
Control system	60 ± 20	65 ± 20
Air inlet assembly	20 ± 5	20 ± 5
Reduction drive assembly	30 ± 10	35 ± 10
Auxiliaries	35	40
Total engine ready-to-run	\$630 ± 75	\$735 ± 85

A particularly key entry in Table 7-14 is the cost of production machinery for the free-turbine Brayton engine. This cost is more than twice that for the Otto engines. Among the major reasons for the large machinery investment for the Brayton engine is that metal cutting speeds for superalloys are much slower than for other metals. Other reasons include the close tolerances required and the large number of precision machined surfaces. To meet a required output rate of 400,000 engines per year, several parallel transfer lines are required.

The entries for (1) direct production machinery and (3) foundry and forge capacity from Table 7-14 are summed to give foundry and machinery costs for each of the engines in the first column of Table 7-15. Similarly, the Table 7-14 entries for (2) plant costs and (4) tooling startup costs are summed to provide tooling and plant costs for alternative engines in the second column of Table 7-15. These two entries are separated to permit the use of a different economic lifetime for the foundry and machinery than for tooling startup and plants.

Table 7-11. Variable Costs for a Metallic Swashplate Stirling Engine

Major Component	Cost in 1977\$	
	100 HP	150 HP
Heater head assembly	175 ± 70 ⁽¹⁾	260 ± 75 ⁽¹⁾
Combustor subassembly	6 ± 5	8 ± 5
Top cover	4 ± 2	5 ± 3
Cycle regenerator and cooler	15 ± 5	18 ± 5
Piston assembly	50 ± 15	60 ± 15
Rod seal assembly	60 ± 15	60 ± 15
Swashplate	10 ± 5	12 ± 5
Main shaft	10 ± 5	12 ± 5
Cylinder block	75 ± 10	90 ± 10
Combustion auxiliaries	30 ± 10	35 ± 10
Preheater assembly	80 ± 10	90 ± 10
Coolant pump and drive	30 ± 10	35 ± 10
Power control system	60 ± 25	60 ± 25
Fuel supply controls	25 ± 5	25 ± 5
Accessory drive system	33 ± 10	35 ± 10
Engine Auxiliaries	40	40
Radiator assembly	44 ± 5	55 ± 5
Totals	\$747 ± 80	\$900 ± 85

(1) Considerable uncertainty regarding the ultimate design of the heater head assembly gives rise to the large error bound.

Table 7-12. Direct Labor Hours by Labor Category

Engine Type	Foundry	Machining	Assembly/Test	Total
Brayton Free-Turbine	4	4	2	10
Stirling	6	1	3	10
Otto 2-way catalyst 100 hp	6	2	1	9
Otto 2-way catalyst 150 hp	6	3	1	10
Otto 3-way catalyst 100 hp	6	2	2	10
Otto 2-way catalyst 150 hp	6	3	2	11

7.10 RETURN ON INVESTMENT

The money invested in production facilities must be recovered through sales of engines. If it is assumed that an equal amount C is to be recovered each year during the economic lifetime t of the production facilities and that the quantity recovered is to provide a rate of return R on the original investment I , it can be shown that

$$C = I \frac{R}{1 - (1+R)^{-t}}$$

In simple terms, C provides for recovery of capital as well as return on investment. To prorate C to individual engines, it is necessary to divide by the annual production volume. Assuming a 15 percent rate of return, a 10-year lifetime for foundry and machining, a 20-year lifetime for plants and startup tooling, and a 400,000 annual engine production volume, the investment recovery per unit produced (\$/engine) can be as presented in Table 7-15. No investment credit is taken on income taxes.

7.11 COST BREAKDOWN

The various contributors to the total cost of producing the different engines are given in Table 7-16. The variable costs in Table 7-16 are taken from the material and labor variable costs listed in Table 7-12. Special tooling costs are presumed to vary proportionally with variable costs. Using data for gasoline engines with oxidation catalysts, special tooling was found to be 7.5 percent of variable costs. This same proportion was then applied to the other engines.

Table 7-13. Variable Costs of Conventional and Advanced Automotive Engines

Material Type	Brayton Engine (metal)				Stirling Engine (metal)				Carbureted Gasoline Engine Oxidation Catalyst				Fuel-Injected Gasoline Engine 3-Way Catalyst			
	100 HP (4)		150 HP (4)		100 HP (4)		150 HP (4)		100 HP (4)		150 HP (4)		100 HP (4)		150 HP (4)	
	Wt. lb	Cost \$	Wt. lb	Cost \$	Wt. lb	Cost \$	Wt. lb	Cost \$	Wt. lb	Cost \$	Wt. lb	Cost \$	Wt. lb	Cost \$	Wt. lb	Cost \$
Cast iron	145	29	173	35	69	14	87	17	180	36	250	50	200	40	270	54
Carbon steel	-	-	-	-	116.3	24	114.8	30	180	38	250	53	180	38	250	53
Alloy steel	20.5	4.3	22.7	5.7	14	4	18	5	5	1	5	1	5	1	5	1
Austenitic	22.5	13.5	24.2	14.5	67.5	41	84	51								
Precipitation hardening stainless	6.8	10.9	7.0	11.2												
Superalloy	14.4	64.8	16.1	72.5	11.8	53	15	61								
Ceramic	19	23.8	20.	25.	21	26	26	33								
Aluminum alloy	2	1.3	2	1.3	158	103	199	129	18	12	20	13	18	19	20	13
Copper alloy	-	-	-	-					20	19	20	19	20	19	20	19
Miscellaneous (1)					248.2	425	271	462	90	222	125	269	90	230	125	272
Variable (2) Material	427.5	571	477	631	706	690	845	795	493	328	670	405	513	340	690	412
Variable (3) labor (hrs) & \$	(10)	80	(10)	80	(10)	80	(10)	80	(9)	72	(10)	80	(10)	80	(11)	88
Total Variable Cost		651		711		770		875		400		485		420		500

(1) Not broken down (conventional auto materials) — nonhomogeneous, miscellaneous, and purchased parts.

(2) Not including overhead.

(3) Includes foundry labor and uses an average labor rate of \$8.00 per hour.

(4) These are finished part weights, without scrap allowance, since loss of scrap, that is not recycled, is assumed to be small.

Table 7-14. Engine Production Facility (in Millions of 1977\$) — Sized to Produce 400,000 Units per Year

Cost Category	Engine Type			
	Brayton Free Turbine	Stirling	Present Gasoline Otto	Advanced Gasoline Otto
1. Direct production machinery	250	175	85	110
2. Plant cost	130	120	85	95
3. Foundry and forge capacity				
A. Cast iron and steel forgings	20	15	60	70
B. Superalloys and stainless steel	60	60	0 ⁽¹⁾	0 ⁽¹⁾
C. Aluminum	Small	75	6	6
4. Tooling costs, tooling setup, and vendor tooling	<u>65</u>	<u>65</u>	<u>60</u>	<u>60</u>
Total Costs	525	510	296	341

(1) Stainless steel in catalyst container not included.

Factory fixed costs involve a myriad of items from paper towels to utilities for heating and cooling the workspace. Although these costs are largely independent of engine cost and volume, there is some variation in this cost with the variable cost, e.g., more workers, more paper towels used. In Table 7-16, factory fixed cost was taken to be \$200 plus 10 percent of the variable cost for each engine. The two parameters were based on gasoline engine factory fixed cost estimation.

Manufacturing cost for each engine was defined to be the sum of variable cost, special tooling, and factory fixed cost. These are presented in Table 7-16.

Corporate overhead must also be allocated to each engine. These costs involve the operation of corporate offices that render a wide variety of service including R&D, design, finance, marketing, and interfacing with government. Although these corporate expenditures

Table 7-15. Facilities, Tooling Amortization, and Investment Costs of Alternative Engines

Engine	Cost in Millions of 1977\$			1977\$ per Unit		
	Foundry and Tooling	Startup and Facilities	Total	Tooling Investment Recovery ⁽¹⁾	Facilities Investment Recovery ⁽¹⁾	Total Investment Recovery ⁽¹⁾
Brayton free-turbine	330	195	525	165	80	245
Stirling	325	185	510	165	75	240
Gasoline 2-way catalyst	151	145	296	75	60	135
Gasoline 3-way catalyst	186	155	341	95	65	160

(1) Assumes a 10-year life for tooling, 20 years for facilities, and a 15 percent rate of return. Production volume is anticipated to be 400,000 engines per year.

are largely fixed, there is some variation with engine variable cost. In Table 7-16 corporate overhead was taken to be \$150 plus 15 percent of the variable cost for each engine. The parameters selected were derived from conventional gasoline engine corporate overhead estimates.

The return on investment for each engine includes that for foundry, tooling, and plant investments as developed in Table 7-15.

The wholesale (OEM) price can be calculated by adding the manufacturing cost, corporate overhead, and return on investment for each engine. Wholesale (OEM) prices for the conventional gasoline, free-turbine Brayton and Stirling engines are given in Table 7-16. The dealer price is obtained from the wholesale price by adding a mark-up of 33 percent to account for dealer profit and other expenses.

7.12 COMPARISONS OF ENGINE VARIABLE COST AND SELLING PRICE

Variable cost and selling price for the various engines are compared in Table 7-17. From the table it is seen that the Brayton and Stirling engines have considerably higher variable costs than the Otto

Table 7-16. Cost Breakdown for Various Engines (1977\$)

Engine	Variable Cost(1)	Special Tooling(2)	Factory Fixed Cost(3)	Manufacturing Cost(4)	Corporate Overhead(5)	Return on Investment(6)	Wholesale Price(7) (OEM)	Dealer Selling Price(8)
Brayton free-turbine, 100 hp	651	49	265	965	295	245	1505	2002
Brayton free-turbine, 150 hp	711	53	271	1035	305	245	1585	2108
Stirling, 100 hp	770	58	277	1105	315	240	1660	2208
Stirling, 150 hp	875	66	288	1229	335	240	1804	2399
Gasoline, 2-way cat., 100 hp	400	30	240	670	250	135	1055	1403
Gasoline, 2-way cat., 150 hp	485	36	248	769	265	135	1169	1555
Gasoline, 3-way cat., 100 hp	420	32	242	694	254	160	1108	1474
Gasoline, 3-way cat., 150 hp	500	38	250	788	268	160	1216	1617

(1) Variable costs based on estimate by type of material from Table 7-12.

(2) Special tooling = 0.075 times variable cost.

(3) Factory fixed cost = \$200 + 0.10 times variable cost.

(4) Manufacturing cost = variable cost + special tooling + factory fixed cost

(5) Corporate overhead = \$150 + 0.15 times manufacturing cost.

(6) Return on investment = tooling investment recovery + facilities investment recovery from Table 7-15.

(7) Wholesale price = manufacturing cost + corporate overhead + return on investment.

(8) Dealer selling price = 1.33 times wholesale price (this gives dealer 25 percent for profit and other expenses).

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Table 7-17. Variable Cost and Selling Price of Various Engines

Engine	Variable Cost in \$	% Increase in Variable Cost Over 2-Way Otto	Selling Price in \$	% Increase in Selling Price Over 2-Way Otto
Brayton free-turbine, 100 hp	651	63	2002	43
Brayton free-turbine, 150 hp	711	64	2108	36
Stirling, 100 hp	770	93	2208	57
Stirling, 150 hp	875	80	2399	54
Otto cycle, 2-way cat., 100 hp	400	-	1403	-
Otto cycle, 2-way cat. 150 hp	485	-	1555	-
Otto cycle, 3-way cat. 100 hp	420	5	1474	5
Otto cycle 3-way cat., 150 hp	500	3	1617	4

cycle engines, but when comparisons of the selling prices are made, the percentage differences are significantly less because of several fixed components in the selling prices. Graphic illustrations of the cost and selling price differences for these engines are given in Figures 7-1 and 7-2.

7.13 CAPITAL TO OUTPUT RATIOS

An important factor in business economics is the ratio of the capital investment required to produce a product to the value of annual output. This ratio gives an indication of the risk in a venture because the higher the ratio, the more capital-intensive the venture, and the greater the losses would be if a large market did not materialize.

The capital investment to output value ratios shown in Table 7-18 were determined by dividing the direct production machinery costs from Table 7-14 by the product of the sales prices from Table 7-16 with the assumed production volume of 400,000 engines per year. Ratios

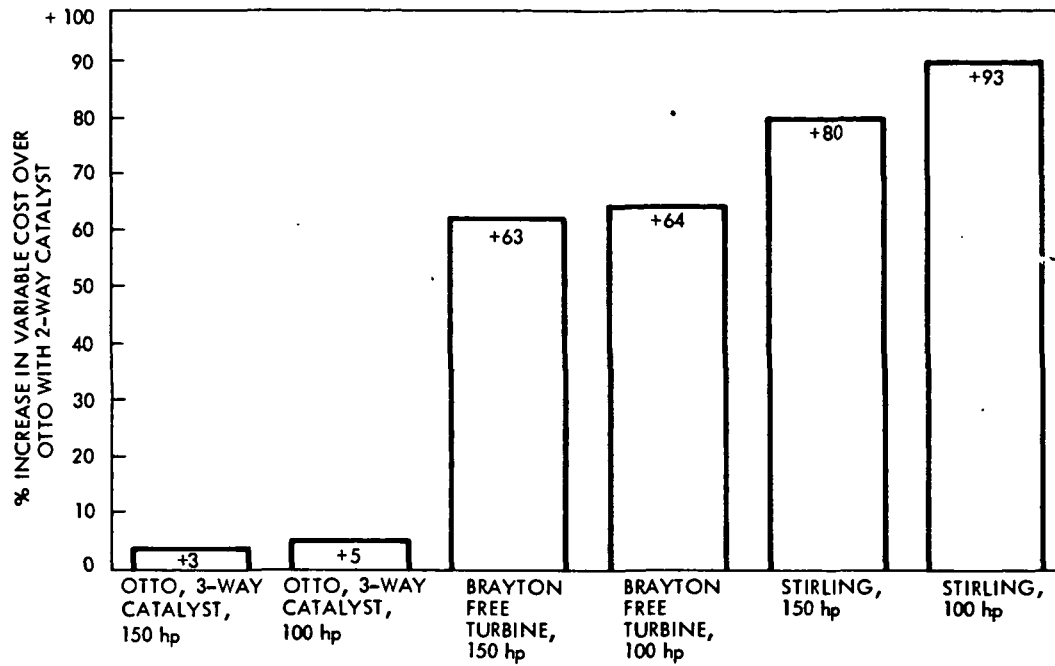


Figure 7-1. Variable Cost Comparisons

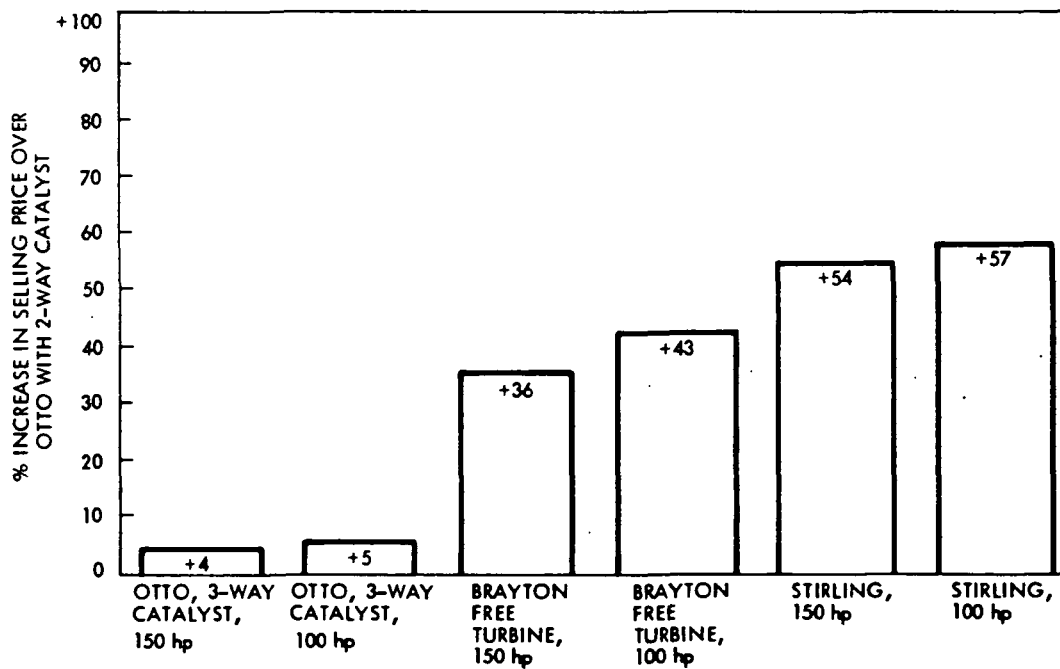


Figure 7-2. Selling Price Comparisons

Table 7-18. Capital/Output Ratios for Various Engines

Engine	Sales Price in \$	Direct Production Machinery (1) in Millions	Capital/ Output Ratio(2)
Free-turbine Brayton, 100 hp	2202	250	0.31
Free-turbine Brayton, 150 hp	2108	250	0.30
Stirling, 100 hp	2208	175	0.20
Stirling, 150 hp	2399	175	0.18
Otto, 2-way catalyst 100 hp	1403	85	0.15
Otto, 2-way catalyst, 150 hp	1555	85	0.14
Otto, 3-way catalyst, 100 hp	1474	110	0.19
Otto, 3-way catalyst, 150 hp	1617	110	0.17

(1) From Table 7-14.

(2) Direct production machinery in \$ million/(400,000 times sales price).

for the gasoline engines vary from 0.14 to 0.19; the Stirling engine ratios are just slightly higher. Capital output ratios for the free-turbine Brayton engine are substantially higher, 0.30 to 0.31, due to the additional lines required to meet a schedule output of 400,000 engines per year.

7.14 SUMMARY OF ENGINE COST RESULTS

The engine cost results are summarized in Table 7-19. Results are given for the selling price, price difference, and price increase factor for the various conventional and advanced engines. The baseline engine is taken to be the conventional carbureted gasoline engine with an oxidation (two-way) catalyst. Because of the many assumptions which must be made in any cost analysis, it is likely that the price

Table 7-19. Summary of Engine Cost Results

Engine Type	Selling Price, \$	Price Difference, \$	Price Increase Factor
Gasoline, carbureted, oxidation catalyst			
100 hp	1403	0	1.0
150 hp	1555	0	1.0
Gasoline, fuel-injected, 3-way catalyst			
100 hp	1474	71	1.05
150 hp	1617	62	1.04
Diesel, turbocharged ⁽¹⁾			
150 hp	1786	169	1.13
Free-turbine Brayton, metallic			
100 hp	2002	599	1.43
150 hp	2108	553	1.36
Stirling, metallic			
100 hp	2208	805	1.57
150 hp	2399	844	1.54

⁽¹⁾ Diesel cost taken from Reference 7-2 using a 20% inflation factor.

difference and price increase factor values are more meaningful than the absolute selling prices indicated. Cost results are given in Table 7-19 for both 100 hp and 150 hp engines of each type. Extrapolation of the results to smaller 50-60 hp engines is probably not valid because the cost of the 100 hp engine was itself already inferred from that of the 150 hp engine.

The free-turbine Brayton and Stirling engine results given in the table are limited to engines using metallic components except for a ceramic core regenerator or air preheater. Sufficient information on manufacturing processes with structural ceramics is not yet available to permit the costing of ceramic engine components such as a turbine wheel or Stirling engine heater head. In addition, the present costing study did not include consideration of the single-shaft Brayton engine, which was found in Reference 7-2 to be about 15 percent lower in cost than the two-shaft configuration which was considered in the present work.

The engine cost results given in Table 7-19 indicate that both the free-turbine Brayton and Stirling engines would be higher in cost than the conventional gasoline engine at the same horsepower. The projected price increase factor would be about 1.4 for the Brayton engine and 1.55 for the Stirling engine. In the case of the Brayton engine, the price increase factor will be slightly lower than 1.4 if the engines are compared on the basis of equivalent performance in a vehicle because of the better specific weight and advantageous torque characteristics of the free-turbine Brayton engine. For example, if a 125 hp gasoline engine is compared with a 100 hp free-turbine Brayton engine, the estimated price increase factor would be 1.33 rather than 1.4. Further reductions in the price increase factor for the Brayton engine could result from the use of the single-shaft rather than the two-shaft configuration and the use of ceramics rather than superalloys for the turbine and combustor liner components. The magnitudes of the resultant cost reductions are very uncertain at the present time as they are dependent on technology advances in ceramic manufacturing processes and continuously variable transmissions. Even without these possible cost reductions for the Brayton engine, the estimated price difference between the Brayton engine and the conventional gasoline engine is only \$500 - \$600 (1977 dollars) for a full-sized car. This difference is only about 10 percent of the current selling price of that size of car and does not seem to preclude, on an economic basis, the introduction of Brayton engines in such passenger cars. The cost difference between the Brayton and the diesel is estimated to be \$300-400.

As indicated in Table 7-19, the cost difference between the Stirling engine and the conventional gasoline engine is estimated to be about \$800 (1977 dollars) for engines in the 100-150 hp size range. It is likely that for equivalent vehicle performance, the size (hp) of the Stirling and gasoline engines will be the same and differences in required engine size will not affect the engine cost situation as in the case of the Brayton engine. In addition, the use of ceramics is likely to have a smaller effect on cost than it has for the Brayton engine since the cost of the superalloy heater head is a small fraction of the total engine cost. The relatively large cost differential for the Stirling engine will thus need to be justified by improved fuel economy, lower emissions, and greater fuel versatility. In this sense, the Stirling engine situation is comparable to that of the diesel engine which is currently being introduced at a relatively large cost premium. Even though the initial cost picture for the Stirling engine is currently less favorable than for the Brayton engine, it does not seem to preclude the eventual introduction of the engine if its excellent fuel economy potential can be realized.

NOTE

Fact-finding visits by ATSP team members in connection with the subject matter of this section were made as follows:

Chrysler Corporation, Highland Park MI	July 1977
Ford Motor Company, Dearborn MI	July 1977
Williams Research Corporation, Walled Lake MI	July 1977
Garrett Corporation, Los Angeles CA	July 1977
Audi NSU Auto Union, Neckarsulm, West Germany	Sept 1977
Daimler-Benz, Stuttgart, West Germany	Sept 1977
Ricardo & Company Engineers, Ltd., Shoreham-by-Sea, England	Sept 1977
Philips Research Laboratories, Eindhoven, The Netherlands	Sept 1977
MAN, Nuremberg, West Germany	Sept 1977
Saab-Scandia, Trollhattan, Sweden	Sept 1977
British Leyland, Coventry, England	Sept 1977

SECTION 8

CERAMIC TECHNOLOGY FOR HEAT ENGINE APPLICATIONS

8.1 INTRODUCTION

The role of materials technology for vehicle heat engine application is important. The potential high thermal efficiency of Brayton or Stirling engines will be realized with higher operating temperatures. A fully regenerated Brayton engine operating at 1850°F (1010°C) allows prediction of a cycle thermal efficiency of 26.8 percent. An increase in the operating temperature of 650°F (343°C) theoretically increases the efficiency to 48.3 percent (Reference 8-1). The maximum operating temperature for a given heat engine is dictated by the materials employed. Brittle material heat engine components have the potential of operating at temperatures approaching 2500°F (1371°C).

This section will present a general discussion of the highlights of recent ceramic material and fabrication developments and outline major ceramic technology problem areas related to the metallic and advanced Brayton and Stirling engine configurations. Most of the discussion will be directed toward high temperature Brayton engines; however, the application of ceramics to Stirling engine technology for automobiles will also be addressed. Design analysis studies have shown that high temperature operation of Brayton engines will be required to attain the attractive efficiency which these engines offer. Thus, this discussion will emphasize the state of development and performance of Brayton engine ceramic components which have the capability of operating at temperatures hundreds of degrees above metallic components.

The metallic configuration Brayton and Stirling engines require ceramics only for the regenerator and the preheater, respectively. The advanced configurations are longer-term projections that are predicated on the large scale application of structural ceramics. The efforts of the present Automotive Technology Status and Projections Projects (ATSP) to assess ceramic materials technology started with the data base from the Automotive Power Systems Evaluation Study Report (Reference 8-2). Comments and critiques from interested parties (Reference 8-2) in the automotive community were carefully considered. Numerous fact-finding visits were conducted to both ceramic manufacturers and engine developers. (1) The state-of-the-art of ceramic materials technology as determined from the open technical literature was combined with the preceding information in preparing the assessments contained in this section.

(1) For a list of visits to manufacturers and developers, see Note at end of this section.

Historically, ceramic materials have been used in refractory applications where their high melting temperature and insulating properties are of prime importance. Their load-carrying capabilities have usually been marginal. Only recently have design applications been considered in which the high temperature structural performance of ceramic materials was the critical design parameter. The DOE/ARPA/Ford ceramic turbine engine program has resulted in great strides in high-performance ceramic technology. However, additional advancement in the technology is required. At present, ceramic material development and fabrication process development for ceramic materials is in a state of rapid change and improvement. Supporting technologies such as brittle material design techniques, nondestructive evaluation, techniques for mechanical testing, and quality control are likewise in a state of development. It would be presumptuous to predict the ultimate state of the art, but the demonstrated properties of such materials in high stress, high temperature applications and the rate of improvement in these properties are encouraging.

The development of ceramic heat engine components which demonstrate engineering feasibility along with massproducibility at a reasonable cost rests on the successful interaction of several technical disciplines. The load-carrying ability and reliability of brittle materials are governed by and sensitive to each individual manufacturing process, from raw powder preparation through green body forming, firing, and machining. Ceramic materials lack ductility, and this complicates the component design. To insure the development of a structurally sound and thermodynamically efficient design, designers must consider the appropriate material properties along with the failure criteria and the statistical failure nature of brittle materials. The development of materials which demonstrate the desired high strengths at high temperature along with ease of fabrication is essential.

8.2.1 Material Development

Ceramic materials which demonstrate high load-carrying capabilities beyond simple compressive loading are referred to as structural ceramics. These materials are categorized into families of ceramics designated as borides, carbides, nitrides, oxides, and others. Boride ceramics, compounds of boron with metals, usually exhibit good oxidation resistance at elevated temperatures. The refractory diborides demonstrate strength retention along with oxidation resistance at elevated temperatures. Commercial applications for borides are limited; there is no significant production of boride ceramics, and they are not being considered for automotive applications. Carbides, compounds of elements with carbon, usually have high melting temperatures and are extremely hard. Lack of oxidation resistance is a general limitation of carbides; however, silicon carbide (SiC) is a notable exception. A variety of silicon carbide products are commercially available and are used in high temperature oxidizing environments. Nitrides, compounds of elements with nitrogen, exhibit considerable variation in properties between members of this material family. Silicon nitride is the most readily

available of the refractory nitrides. Silicon nitride (Si_3N_4) has demonstrated high strength capabilities with good oxidation resistance at elevated temperatures. Since oxides occur naturally, they are widely utilized, readily available, and well characterized. The primary asset of oxides is their resistance to oxidation; however, their maximum temperature capability is lower than that of SiC or Si_3N_4 in structural applications. Information on the detailed material properties of specific ceramic materials is available in Reference 8-3.

Structural ceramic materials which have been and/or are presently under development for heat engine applications due to their high temperature structural potential are silicon nitride, silicon carbide, silicon/silicon carbide, and sialons.

8.2.1.1 Silicon Nitride. Hot-pressed and pressureless sintered silicon nitride demonstrate high strength capabilities. The former has greater strength because it has higher density due to the fabrication technique employed. Both materials require sintering aids such as magnesium oxide to promote densification. Unfortunately, these sintering aids affect the high temperature behavior of the material, resulting in a reduction in high temperature strength. The formation of glassy phases in the grain boundaries leads to the reduced strength, owing to viscous shear (Reference 8-4). Development work has been successful in identifying sintering aids superior to magnesium oxide. For example, yttria additions to hot-pressed silicon nitride have produced material with greatly enhanced high temperature properties (References 8-5 to 8-7).

Silicon nitride powder technology is still in the development state. Obtaining consistent, uniform powders with the necessary characteristics is an unresolved problem. Batch variations in the properties of powders from current suppliers can result in a lack of consistency in the properties of finished parts. The lack of a large volume supplier of suitable powder is not surprising. It should be recognized that there is, as yet, no large market for such material; consequently, there is no economic motivation to develop such a supply.

Reaction-sintered silicon nitride is produced by compacting finely divided silicon powder into the desired configuration (Reference 8-8). Conventional ceramic or plastic forming techniques such as slip casting or injection molding are utilized to form the green body. The silicon compact is then subjected to a high temperature nitrogen environment. The chemical reaction between silicon and nitrogen yields the desired silicon nitride form with almost insignificant dimensional change. This form of silicon nitride is not as strong as the high density, hot-pressed or pressureless sintered material, but there are advantages due to versatile fabrication capabilities. In addition, this process eliminates the need for sintering aids which reduce the creep resistance of the material. Therefore, reaction-sintered silicon nitride demonstrates a higher creep resistance than the hot-pressed or pressureless sintered material.

8.2.1.2 Silicon Carbide. The development of silicon carbide materials has been pursued by several organizations. Significant progress has been made in the areas of sinterable powders, sintering aids which do not drastically reduce the high temperature properties, and in forming and sintering techniques which produce parts very near to the final shape and dimensions. The high strength, coupled with good oxidation and creep resistance of the best SiC produced, make it an important candidate material for high temperature mechanical components for heat engine applications. Silicon carbide is available in hot-pressed, reaction-bonded, and solid state-sintered forms.

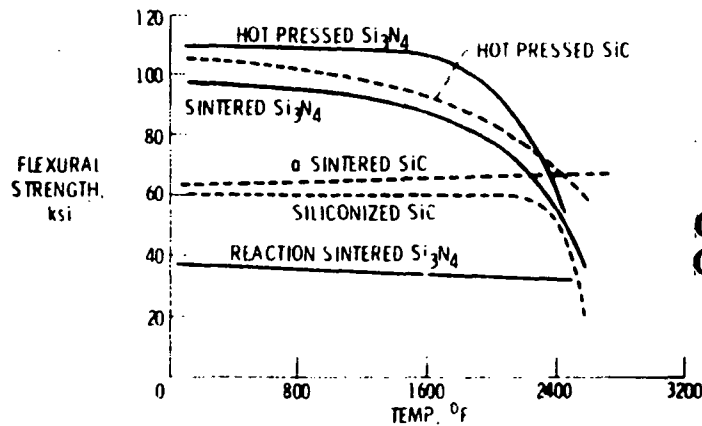
Substantial reductions in the required pressure and temperatures to obtain fully dense hot-pressed SiC have been realized by the development of sintering aids (Reference 8-9). Minor amounts of these selected impurities not only serve to reduce the fabrication pressure and temperature, but also minimize grain growth (Reference 8-10). The fine grain size and dense microstructure of hot-pressed SiC lead to its high strength at both room and elevated temperature.

Reaction-bonded silicon carbide, as the name implies, is produced by utilizing a reaction of carbon and silicon to bond existing SiC grains together. Silicon carbide powder is mixed with carbon and an organic binder and formed into the appropriate configuration. Sintering is accomplished in a liquid silicon environment whereby the liquid silicon diffuses into the body and reacts with the carbon. The resultant material consists of SiC from the reaction and the original powder along with regions of free silicon. The strength of reaction-bonded SiC, which is less than that of hot-pressed material, has a high temperature limitation of approximately 1200°C. Near the melting point of silicon considerable plasticity occurs with a corresponding decrease in strength.

To overcome the fabrication disadvantages related to hot-pressing while retaining the required high densities for high strength, pressureless solid state sintering techniques for SiC are under development. These techniques utilize an activating agent to sinter SiC powder to densities exceeding 98 percent of the material's theoretical density (Reference 8-11). Solid state-sintered SiC does not demonstrate the reduction in strength at elevated temperature that is observed in the reaction-bonded material; however, shrinkage on the order of 18 percent during the solid state sintering process is expected. Precise control of mixing, forming, and firing are therefore required to minimize nonuniform shrinkage, thus avoiding reductions in the component strength and reducing the required final machining to obtain desired dimensional tolerances.

Figure 8-1 displays the strength of several SiC and Si₃N₄ materials as a function of temperature (Reference 8-12).

8.2.1.3 Silicon/Silicon Carbide. Silicon/silicon carbide composite materials consist of silicon carbide fibers separated by silicon metal filler. Carbon fibers are reacted with silicon at a temperature of 2700°F to form silicon carbide fibers within the molten silicon. The



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Figure 8-1. Strength of Several SiC and Si_3N_4 Materials (Reference 8-12)

composite can then be precision cast with little or no shrinkage. If machining is required, it can be accomplished utilizing diamond cutting tools.

Because of the metallic silicon matrix, the composite demonstrates greater toughness than the previously discussed ceramic materials. The room and elevated temperature strength is dependent upon the volume percent of SiC fibers (Reference 8-13). Increasing amounts of SiC increase the strength. The temperature capability of the composition is dictated by the silicon matrix, which limits this material family to lower temperatures than SiC or Si_3N_4 . A sharp decrease in the strength is realized at temperatures which approach the melting point of silicon. The bend strength of several General Electric silicon/silicon carbide composites as a function of temperature is presented in Figure 8-2 (Reference 8-14). Fiber orientation can be either random or aligned. Aligned fiber composite materials exhibit anisotropic properties. The maximum strength of Si/SiC composite is in the direction parallel to the SiC fibers. Therefore, the fiber orientation must be considered from a design and fabrication standpoint.

8.2.1.4 Sialons. Ceramic materials consisting of an intimate mixture of silicon nitride and aluminum oxide are usually designated as sialons. It has been suggested that a pressureless sintered sialon material could possibly demonstrate properties approaching those of hot-pressed silicon nitride (Reference 8-15). The fabrication advantages of pressureless sintering over hot-pressing has prompted the investigation of sialons. The Ford Motor Company conducted a program which included material composition and sintering techniques in hopes of developing a pressureless sintered sialon with the required high temperature properties for Brayton engine components (Reference 8-16). Sialons, with compositions of 16 to 40 percent alumina, 1 to 25 percent yttria, and the remaining balance of silicon nitride, have been investigated. These compositions have been sintered in argon as well as nitrogen

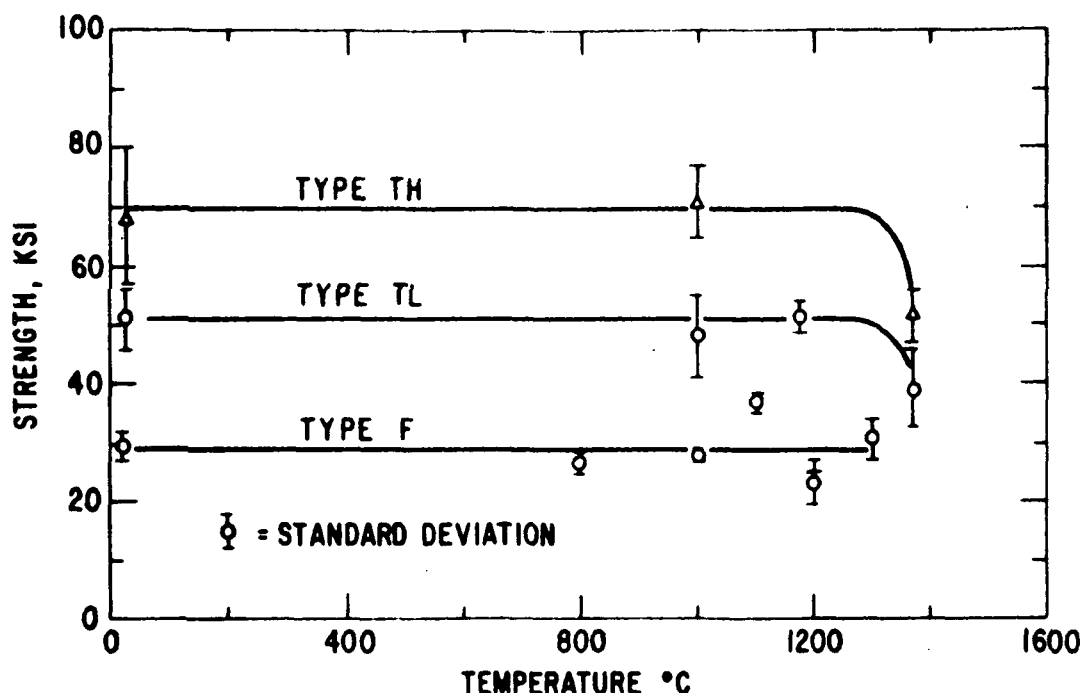


Figure 8-2. Bend Strength of Several Si/SiC Materials (Reference 8-14)

atmospheres at temperatures between 1600 and 1700°C at times from 4 to 12 hours.

Because of the promising room temperature strength of several sialons, a high temperature strength testing program of a few materials was instigated. Some of the results obtained by Ford at room temperature and elevated temperatures on several sialons are summarized in Figure 8-3 (Reference 8-16). The test data indicate a substantial reduction in strength at elevated temperatures. This reduction in strength as well as material X-ray analysis suggests the presence of a glassy phase (Reference 8-17). Data to date would seem to indicate that the application of sialons for 2500°F (1731°C) heat engine components is unlikely because of its reduced high temperature strength.

8.2.2 Fabrication Methods

The properties of ceramics which impart their outstanding high temperature performance also result in the materials being difficult to fabricate. Owing to a lack of ductility, traditional metal-working techniques such as forging and rolling are not applicable. The mechanical strength of a given ceramic material is dependent upon its density, among other things, which in turn is dependent upon the fabrication technique employed. The closer the material is to its full density, the greater the material strength. Unfortunately, to achieve greater densities, increasing fabrication difficulties are encountered. To reduce costly ceramic machining, the ceramic body must be formed and then sintered to the approximate final dimensions.

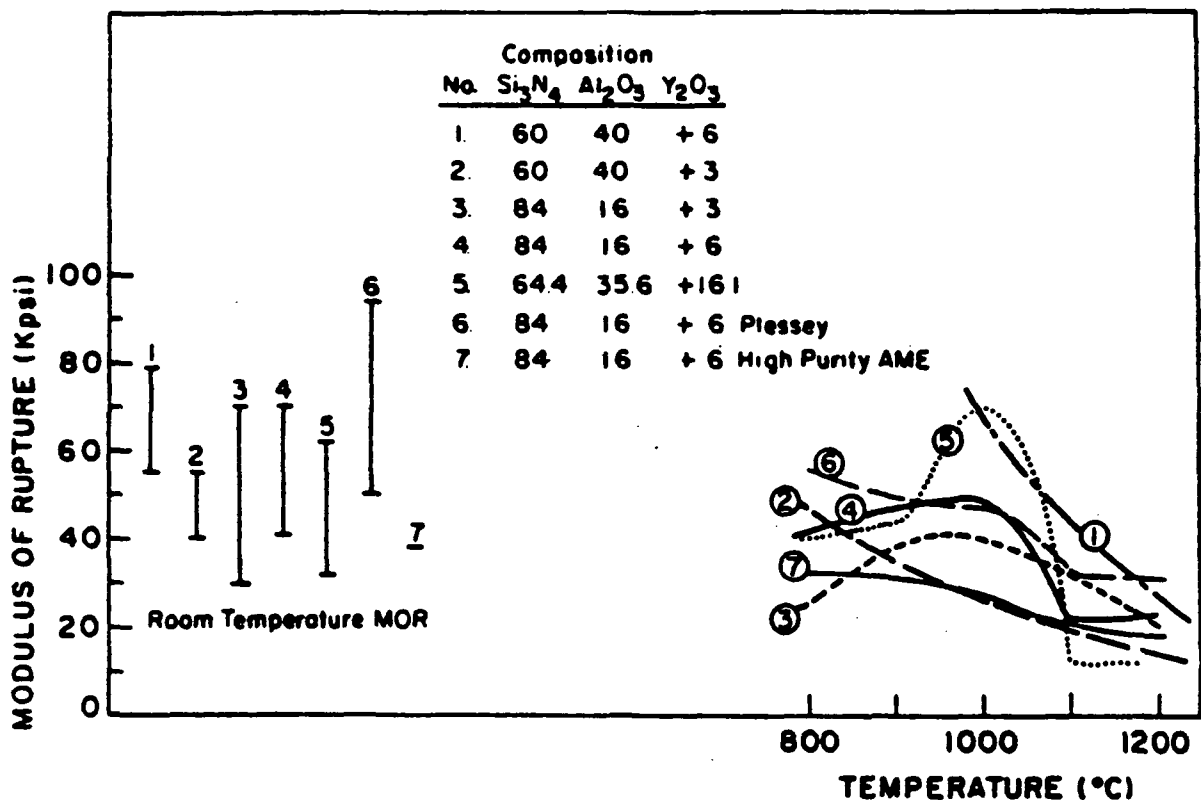


Figure 8-3. Effect of Temperature on Modulus of Rupture for Several Sialon Materials (Reference 8-16)

Usually, three fabrication stages are required to produce a structural ceramic body from the processed raw material: forming, firing, and machining. The material is formed as a powder/binder mixture into the "green" configuration. After forming, the part is fired (sintered) at elevated temperatures to effect the diffusion between the particles which results in densification of the body. The part can then be machined as required to final dimensions.

8.2.2.1 Forming. Various methods are used to form the green, unsintered body, e.g., slip casting, injection molding, isostatic pressing, and extrusion.

The slip cast forming process involves mixing appropriate sized ceramic particles with a suspending fluid, typically water, into a slurry (Reference 8-18). The slurry is then poured into a porous plaster mold which draws the fluid from the slurry, producing a hardened ceramic layer at the mold slurry interface. This process can continue until the entire mold is filled, or the mold can be drained leaving a

ceramic body of suitable wall thickness. Part geometry in the slip casting process is limited to that which can be formed in conventional plaster tooling.

Ford Motor Company utilized the slip casting process to form reaction-sintered silicon nitride turbine rotor blade rings; however, problems were encountered during the nitriding cycle owing to the green ware density (Reference 8-16). Injection molding is presently being employed, producing a lower density body but eliminating nitriding cycle problems.

Injection molding appears to be a very promising technique for forming complex shapes such as turbine rotor blades, stators, and nose cones; however, owing to the limiting of densities achievable with this process, a reduced material strength is realized. This process utilizes a mixture of raw material particles, and an organic binder. This mixture is then forced under pressure into the appropriate die. The binder holds the body together and acts as a die lubrication agent in the forming process. Subsequently, the binder is burned off during the sintering process.

Isostatic pressing or cold pressing is a process whereby a dry or slightly damp ceramic powder is compacted into dies at sufficiently high pressure to form a dense and strong "green" body. Technical ceramics are typically pressed at pressures in the order of 8000 psi which minimizes the firing shrinkage to within 1 percent. Size and geometric configuration are the limiting factors in the isostatic pressing process. Compaction pressure variation due to complex die configuration or from wall friction in long die lengths cause warping and defects during firing.

Silicon powder can be isostatically pressed into rough shapes as required and then machined to the final configuration before the part is nitrided. This is a costly approach, but it lends itself to prototype fabrication. Shrinkage in the nitriding step is generally less than 0.1 percent with densities in the range of 2.3 to 2.7 g/cc (Reference 8-19).

8.2.2.2 Sintering. Once the ceramic body is formed into the desired configuration it is fired to sinter the ceramic particles together. Unlike metallic materials, the ceramic material does not melt and then solidify. Instead, the particles, which are in intimate contact with each other, transfer material by evaporation-condensation and solid state diffusion. To insure a successful sintering process, the initial ceramic particle size and sintering temperature must be well controlled.

Closely related to the problem of sinterability is the production of good starting powders. Powders must be carefully and uniformly prepared to have a desirable particle size distribution and controlled impurity levels. At the present time there are no manufacturers of silicon nitride powders who can supply quality powder in the volumes required for the mass production of automobile components. However, increasing demand for silicon nitride will result in the development of an adequate supply of powder. Silicon carbide powder enjoys the

advantage of being manufactured by the Atchenson process. This technique permits high volume production. Adaptation of the Atchenson process to produce SiC powder of the quality required to mass produce automotive components is under intensive development.

Because of the refractory nature of ceramic materials, sintering temperatures are quite high. To aid diffusion during sintering, materials are added to the powder as "sintering aids". These materials (e.g., MgO in silicon nitride and boron in silicon carbide) promote sintering by increasing the mobility of the diffusing species and result in lower sintering temperatures and shorter sintering times. Unfortunately, these sintering aids reduce the high temperature creep resistance of the material. What is required for ultimate high temperature properties is a very pure material (one which contains the sintering aids, if utilized, in concentrations less than the solubility limit) which can still be sintered. By this approach, a second, weakening phase is not formed at the grain boundaries of the sintered body.

The amount of shrinkage and warping encountered during sintering depends upon the green state density and density variations. During densification, the open volumes of pores within the formed ceramic body contract, leading to shrinkage. For complete densification, the fractional porosity present in the green body equals the shrinkage. Density variations cause warping because of localized shrinkage, which in turn can cause cracks to form.

8.2.2.3 Hot-Pressing. A fabrication process which includes forming and sintering simultaneously is hot pressing. This technique utilizes press forming at the required temperature to induce sintering. Ceramic bodies which approach the material theoretical density, and thus exhibit superior mechanical properties, can be obtained from this process. However, the difficulties and limitations encountered in press forming are magnified by the temperature requirements of hot pressing.

The hot pressing technique has proved to be a costly manufacturing approach. The unavailability of inexpensive long life dies required for the high-temperature, high-stress, hot-pressing process, coupled with the difficulties in achieving a high speed production facility, leads to the high cost. Simple shapes can be hot pressed and then machined to required geometric configuration and tolerances - an expensive labor-intensive operation which is not readily adaptable to mass production.

8.2.2.4 Machining. The machining of ceramic components is important from a cost standpoint as well as a technological standpoint. Since it is a slow and labor-intensive operation it often represents a major cost in producing structural ceramic components. The machining process affects the resultant strength and fracture behavior of the component because of the introduction of surface defects.

Traditional ceramic machining techniques are characterized as abrasive removal processes such as grinding, sawing, and lapping or polishing (Reference 8-20). The material removal is accomplished by

chipping or spalling particles from the surface. For hard technical ceramics such as Si_3N_4 or SiC , diamond grinding wheels are required.

Fracture and plastic deformation are encountered during the machining operation. The material tensile strength is exceeded and material is fractured from the surface. In addition, the momentary tension applied at the surface causes cracks to extend down into the ceramic body, and these reduce the material's strength. The machining damage can be minimized by reducing the material removal rate and grinding or lapping with finer grit wheels or lapping paste.

Because few studies have been conducted, the effect of machining on the high temperature strength of ceramic materials is not well understood. It would be reasonable to assume that, for ceramic parts, because of their greater plasticity and crack-healing capability, the effect of machining damage on high temperature strength would be less than for conventional materials.

Material removal techniques other than abrasive processes are available for the machining of ceramics. They include ultrasonic machining, ion beam sputtering, electron beam machining, electrical discharge machining, and others (Reference 8-21). At present, these processes are far too expensive for mass-produced heat engine ceramic components. Some development work is being conducted in hopes of reducing the cost, but major breakthroughs are not anticipated in this area in the near future. This points out the need to avoid machining operations whenever possible.

8.2.3 Design Techniques

The ceramic materials being developed for structural component applications in Brayton and Stirling engines are brittle materials. They do not possess the forgiving ductility which would allow the material to deform and redistribute the stresses in areas where high concentrations of stress exist. Also, the fracture toughness of ceramics is much lower than that of metallic alloys used in such applications. This greatly increases the sensitivity of the strength of ceramics to inherent or introduced flaws in the material. Statistical approaches must be taken to describe the strength of a brittle component in terms of the nature of the stress present in a given volume of material.

Ceramics are considerably stronger under compressive loading than tensile loading. Ratios of compressive to tensile strength have been measured as high as 18 to 1, which may be typical of the range for high performance ceramics (Reference 8-22). Interpreted in terms of component design, this means critical areas should be designed to be in compressive stress states. An alternative approach to requiring increased tensile strengths of ceramic material is to utilize their existing compressive strengths. This approach requires establishment of design criteria much different than that used for ductile materials in which maximum compressive design loads are identical with tensile load limits.

One of the greatest problems to be overcome with respect to the application of brittle materials to structural elements is the tendency of designers to use ductile-materials design approaches and to simply replace the metallic material with a brittle ceramic. The failure criteria for brittle material differ markedly from the Tresca or von Mises criteria for ductile metal. The establishment of viable brittle materials design techniques, such that ceramics are utilized to their utmost potential, must be based on a well defined and verified brittle materials failure criteria.

The design approaches for predicting and improving component survival have received considerable attention recently (Reference 8-23). Finite element stress analysis has been combined with probability of failure statistics which are sensitive to the volume of material subjected to a given stress level (notably Weibull, weakest link statistics). Using this technique, the probability of failure of each volume element of a part may be determined from the given loading conditions, the characteristic strength and Weibull modulus of the material. In addition, attempts are being made to add surface effects and to consider biaxial stress states in the models. These calculations may be used to improve designs by identifying areas where failure is most likely to originate. Such calculations are also being used to predict the probability of survival of whole components. There are problems, however, in trying to glean too much information from a limited input of data. The predictions of most models are highly dependent on the value of the Weibull modulus, which is related to the amount of dispersion in the strength data measured on test specimen material that is assumed to be identical to the material in the component. Large numbers of specimens must be tested to obtain sufficient statistical confidence in the Weibull modulus for predictions of component failure rates on the order of 10^{-3} to 10^{-5} . Methods are in hand to begin quantitatively treating the "inherent" strength variability in ceramics and defining it for design purposes. The capability for producing ceramic material with consistent properties in large quantities remains to be demonstrated.

An alternate or supplemental approach for assuring a given probability of component survival is proof testing. This approach involves overstressing the part beyond its working stress level to cause failure from any flaws which are sufficiently large to propagate cracks. By choosing the overstress to "find" a certain flaw size, the service life may be predicted from the knowledge of the largest flaw present in the material and knowledge of the slow crack growth in the engine environment than currently exists. Proof testing may be the best method of assuring component reliability until nondestructive evaluation methods are developed that can detect critical flaws.

Only the hot flowpath will be constructed entirely from ceramic material. At several interfaces ceramic parts must transmit loads to metallic parts. The location of these interfaces may be chosen by the designer. Mismatch in the modulus and thermal expansion of ceramic and metallic materials creates design problems which are still in need of optimum solutions.

Nondestructive evaluation (NDE) is a method of obtaining information necessary for predicting component service life. Efforts to develop useful NDE techniques for high-performance ceramic materials are under development. The major challenge for developing such techniques is the extremely small flaw size which must be detected. The nature and shape of features on the order of tens of microns in size must be determined since flaws of this size limit the strength of such material. Many techniques are being investigated both from the standpoint of ultimate detectability and application to the mass production environment: acoustic emission, sonic velocity, liquid dye penetrant, ultrasonic scan, X-ray radiography, and others. As yet, no technique has emerged which is capable of detecting or measuring the strength limiting flaws in these ceramic materials.

8.3 STRUCTURAL CERAMIC COMPONENTS

A substantial amount of work dealing with Brayton engine ceramic components has been conducted at the Ford Motor Company with the support of ARPA and DOE. The goal of this program (Brittle Materials Design, High Temperature Gas Turbine) is to develop ceramic components and to demonstrate their structural integrity in a 200-horsepower high temperature vehicular Brayton engine (References 8-15, 8-16, and 8-23). The attainment of this objective will be demonstrated by 200 hours of operation of an uncooled ceramic engine with turbine inlet temperatures of up to 2500°F (1371°C). Additional programs dealing with structural ceramic components for Brayton engine application are the DOE/Detroit Diesel Allison "Improved Heavy Duty Gas Turbine Engine Program" and the ARPA/Garrett Airesearch "Marine Turbine Ceramic Engine Program". The approach taken in the Detroit Diesel Allison program is to increase the operating temperature of a baseline metallic engine by replacing metal components with ceramic ones (Reference 8-24). The final goal is to demonstrate an engine with a turbine inlet temperature of 2265°F (1241°C). The Garrett Airesearch program deals with the development and demonstration of a limited-life ceramic Brayton engine for marine applications.

Structural ceramics also offer a large potential for Stirling engine applications. Stirling technology is under active development by Ford, Philips Research Laboratories, United Stirling, and others. Although little structural test data on ceramic components for Stirling engines has been published, it is apparent that the Stirling heater head offers both the most promise and greatest challenge to ceramic materials developers. Several proprietary Stirling ceramic heater-head component hardware development projects are under way; however, their sponsors consider the state of development premature for widespread public disclosure.

The ceramic hot gas flow components of a Brayton engine can be classified into two general categories: stationary and dynamic. The stationary components include the combustor, nose cone, and stators. The dynamic components are the power and gasifier turbine wheels. Figure 8-4 illustrates these components in a Brayton engine (Reference 8-16).

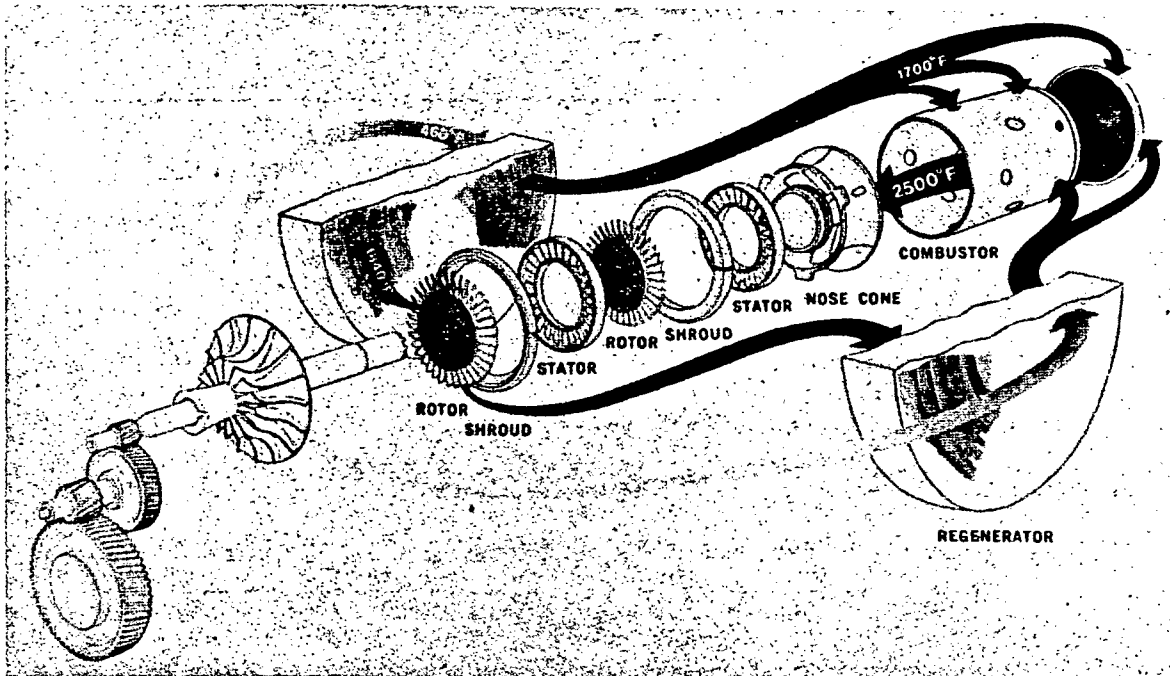


Figure 8-4. Schematic View of a Brayton Engine Flowpath (Reference 8-16)

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8.3.1 Stationary Component Development

The engineering requirements imposed on the stationary ceramic components are those related to the high temperature environment within the engine. The loads induced are usually due to thermal gradients and thermal shock and not to mechanical loading per se. Materials with reduced strength can be used for these components.

The development of stationary ceramic components in the Ford program has proceeded with considerable success. Injection-molded, reaction-sintered silicon nitride nose cones, stators, and tip shrouds have survived limited-life simulated engine tests to temperatures of 2500°F (1371°C) with little deterioration. In addition, reaction-bonded silicon carbide combustors evaluated in a combustor test rig have survived tests to 2500°F (1371°C). Reaction-bonded silicon carbide stators have been tested to 1930°F (1054°C) for 145 hours without failing. A summary of the stationary components tested to date is presented in Table 8-1 (Reference 8-24). These components have demonstrated engineering capability and are expected to satisfy the design requirements during the 200-hour engine test schedule.

A future concern is whether these components can survive 4000 hours of "real world" engine service in vehicular applications.

Table 8-1. Status of Stationary Ceramic Turbine Components (Reference 8-25).

Component	Best Hours of Survival (Without Failure)	
	1930°F	2500°F
Reaction-bonded SiC combustor	175 hours	26 hours
Reaction-sintered Si ₃ N ₄ nose cone	220 hours	5 hours
Reaction-sintered Si ₃ N ₄ stator	175 hours	5 hours
Reaction-bonded SiC stator	145 hours	Not yet tested
Reaction-sintered Si ₃ N ₄ tip shrouds	245 hours	5 hours

8.3.2 Dynamic Component Development

The most critical structural ceramic components in a Brayton engine are the turbine wheels. These components have the most severe design requirements and are the most difficult to fabricate with adequate material properties. Several approaches to fabricating a successful turbine wheel are presently under investigation.

An ARPA-sponsored program is being conducted by the Government Products Division of Pratt & Whitney Aircraft to develop a hybrid ceramic-bladed superalloy disk turbine wheel (References 8-26 and 8-27). The program goal is to develop and demonstrate the attachment of ceramic blades to a wrought superalloy turbine hub. The root section of hot-pressed and diamond-ground silicon nitride blades is sandwiched between two nickel base superalloy disks. The disks are physically bonded together in a forging-diffusion welding cycle. The design incorporates a platinum compliant layer between the blade and disk along with air-cooling passages. A 30-bladed rotor has been fabricated and spun tested for 50 hours at 45,000 rpm, with a blade tip temperature of 2250°F (1232°C) without failure. Additional tests will be conducted on fully bladed rotors up to blade temperatures of 2500°F (1371°C).

Chemical vapor deposition (CVD) of silicon carbide turbine rotors is under investigation by Deposits and Composites Incorporated (Reference 8-28). Fabrications and materials studies have resulted in the production of radial and axial rotors which demonstrate that full rotor components can be produced by CVD techniques. To date, the quality of rotor parts has not reached a level warranting a full scale testing program under engine thermal and mechanical conditions. Cracks and flaws related to residual stress during the CVD growth process represent a serious limitation on the achieved strength of a CVD rotor.

Recent material studies have led to a significant reduction in the number and size of flaws. It is anticipated by Deposits and Composites, Inc. that additional work will eliminate the encountered problems. However, owing to the high cost of CVD techniques, the mass production of CVD turbine rotors is not expected.

The approach taken by the Ford Motor Company, in the DOE program, to develop a ceramic turbine is that of a duo-density silicon nitride turbine design. The turbine blade ring is fabricated from reaction-sintered silicon nitride, which can be formed into complex airfoil shapes by injection molding. The center hub region is hot-pressed silicon nitride. The hub is formed and bonded to the blade ring in a one step hot-pressing operation. Although the reaction-sintered material is of lower strength, it is adequate for the turbine blade ring since the stress levels in this region are lower than in the central hub. In addition, the creep resistance of reaction-sintered silicon nitride is superior to that of the hot-pressed material. The high strength of the hot-pressed material is required in the more highly stressed hub region. Moderate temperatures are encountered in the hub.

Ford recently announced that the first successful hot spin test was completed (Reference 8-29). A duo-density rotor was hot-spun at 50,000 rpm for 25 hours at 2250°F. An additional 1.5 hours at 2500°F (1371°C) was accomplished. The test was terminated when an excessively high temperature was noted in the rotor front face cavity. A gradual shutdown was started by decreasing the turbine inlet temperature while maintaining 50,000 rpm. Failure occurred 6 minutes into shutdown with the turbine inlet temperature at 1900°F (1038°C). Improved fabrications and design contributed to the operation of this test. A significant reduction of flaws in the turbine blade ring has been realized by utilizing an automated controlled injection molding system rather than the previous manually controlled system.

It is interesting to note that the most severe thermal conditions occur during engine burnout or shutdown. Cold air blasts striking hot ceramic components result in the generation of large tensile stresses at the component surface. The surface material, exposed to the cold gas, contracts. The interior hot material constrains the surface contraction, resulting in large thermally-induced tensile stresses. In addition, these stresses are biaxial in nature. The results of a recent mechanical testing program indicate a reduction in the tensile strength of technical ceramics when subjected to a biaxial stress field (Reference 8-30).

The state of ceramic turbine wheel development is encouraging. However, life-cycle tests of prototype turbine wheels are required to demonstrate technical feasibility. The ultimate goal for ceramic turbine wheel fabrication is a monolithic design, which can be mass produced at an acceptable cost. An injection molded, pressureless sintered process seems to be the most promising approach.

The efficiency of Brayton and Stirling engines is greatly improved through the use of heat exchangers, which utilize the high temperature exhaust gases to preheat the incoming air. Preheating is accomplished by a circular rotating open matrix disc heat exchanger which is heated on one half by the exhaust gas and cooled on the other by the incoming air as it is continually rotated. The most likely candidate (Reference 8-1) heat exchangers are referred to as "regenerators" for Brayton engine applications and "preheaters" for Stirling engine applications. Regenerators and preheaters are similar in design and serve the same function in both Brayton and Stirling engines.

The limitations on the regenerator operating temperature and thus its thermal efficiency are dictated by the matrix material. A material is required which exhibits mechanical, dimensional, and chemical stability, coupled with long life at high operating temperatures. Since metallic (stainless steel) regenerators are limited to maximum temperatures of 1200°F (649°C) to 1400°F (760°C), efforts have been directed toward the development of ceramic or "glass ceramic" materials which can operate hundreds of degrees above metallic materials. Glass ceramics are materials which can be formed into appropriate configurations as an amorphous glass material and then transformed into a polycrystalline ceramic during controlled firing.

The major government-funded program dealing with the development of ceramic heat exchangers is the DOE/Ford, "Automotive Gas Turbine Ceramic Regenerator Design and Reliability Program". One of the primary objectives of this program is to develop ceramic regenerator cores that can be used in passenger car Brayton engines, Stirling engines, industrial/truck Brayton engines, and other industrial waste heat recovery systems. The other primary objective is to demonstrate a regenerator life of 10,000 hours at a temperature of 1472°F (800°C) and 3500 hours at 1800°F (982°C) with a failure rate no greater than 10 percent (Reference 8-31). The major ceramic regenerator matrix suppliers are: Corning Glass Company, Coors Porcelain Company, GTE Sylvania, Inc., Owens Illinois Company, and W. R. Grace Company.

Several smaller programs to develop improved ceramic material for regenerator use are: "Owens-Illinois Improved Ceramic Heat Exchange Materials" and "General Electric Improved Ceramic Heat Exchange Materials" (Reference 8-32). These programs are under NASA contract and supported by DOE.

8.4.1 Material Development

During the early stages of regenerator development, Corning fabricated cores utilizing borosilicate glass tubes bonded together by both fusion and silicone-aluminum resin. The discs were evaluated in the Chrysler rotary regenerator test rig, and were found to be satisfactory. Failure occurred at 1400°F (760°C) due to both the high thermal expansion of the glass and the silicon-aluminum cement resin (Reference 8-33).

Corning directed their efforts toward the development of strong, low thermal expansion glass-ceramics, and in 1960 CERCOR^R brand glass-ceramic regenerator cores were introduced. The glass-ceramic material used in these cores was lithium aluminum silicate (LAS). Lithium aluminum silicates of this type are also known as betaspodumenes.

The Ford Motor Company utilized CERCOR^R regenerator cores for their 706 and 707 industrial Brayton engines. This LAS material was found to undergo severe dimensional changes after 300 hours of engine operation. The problem was traced to impurities in the LAS mineral batch. An improved LAS material was developed which resolved the dimensional stability problem due to impurities. It was believed that regenerators fabricated with this improved LAS could demonstrate durability of 4000 to 5000 hours. However, unexpected early failures occurred in service. Test data on 30 regenerators, 11 of which failed, indicated an average life of 1600 hours with a failure rate of 10 percent at 600 hours (Reference 8-31). The LAS material was found to undergo severe chemical attack due to sodium and sulfuric acid.

Sodium can be found in contaminated fuel, salt water, or road salt. In addition, sulfuric acid is formed during the combustion process when sulfur is present in the fuel. Sulfur is present to a greater extent in diesel fuel than in gasoline. The sodium attacks the hot side of the LAS regenerator core, whereas the sulfuric acid condenses on the cold side. Failure occurs because of a bulk material expansion and contraction which occurs within the LAS material owing to an ion exchange process. When LAS undergoes chemical attack, changes in the regenerator core dimensions, coupled with a decrease in the thermal expansion, results in radial cracks within the regenerator core material.

The approach taken by the Ford regenerator program in seeking to improve the performance and durability of glass ceramic core regenerators was to (1) improve regenerator design and (2) develop new ceramic core regenerator materials which have acceptable mechanical and dimensional stability with good resistance to chemical degradation (Reference 6-34).

The improved regenerator design (Reference 8-31) is such that the localized rim stress concentration has been eliminated along with the center hub support load. This configuration utilizes a specially developed elastomer system to bond the core to the outer gear drive ring. The center hub support is eliminated by mounting the regenerator on rollers which run in a center track of the ring gear. The design also incorporates stress-relief slots in the rim of the ceramic core to minimize tangential thermal stresses in this region. The redesign efforts have improved the durability of regenerators fabricated with LAS; however, test results indicate that material improvements as well as these design improvements are required to obtain the needed regenerator life.

A core material with an improved resistance to chemical attack while retaining the low thermal expansion and dimensional

stability of LAS is required. The most promising candidate materials appear to be aluminum silicate (AS) and magnesium aluminum silicate (MAS).

Aluminum silicate is a derivative of lithium aluminum silicate. By chemically leaching LAS in hot sulfuric acid, an ion exchange between lithium and hydrogen occurs, as described earlier. The remaining material, hydrogen aluminum silicate, is then fired at an appropriate temperature to drive out the newly formed water molecules.

The aluminous keatite ceramic material produced by this process has a large negative thermal expansion. However, it is possible to produce an essentially zero average coefficient of expansion over the temperature range of 25-800°C by heating the aluminous keatite to cause a phase transformation (Reference 8-35). The extent of the transformation is controlled to produce a high expansion mullite phase which counterbalances the negative-expanding aluminous keatite phase.

Accelerated chemical attack tests suggest that AS is chemically superior to LAS when exposed to sodium or sulfuric acid. The AS material that demonstrates the best corrosion resistance when exposed to sodium sulfate is keatite. However, keatite has a large thermal expansion, much larger than multiphase keatite and mullite materials (Reference 8-36).

Magnesium aluminum silicate (MAS) materials are also being considered for regenerator cores. The chemical stability of MAS when exposed to sulfuric acid or sodium sulfate is superior to LAS. The magnesium ions in the MAS crystal lattice are more tightly bound than the lithium ions in LAS. This suppresses the ion exchange process between the magnesium and sodium or magnesium and hydrogen ions. The chemical superiority of MAS has been demonstrated under accelerated sodium sulfate and sulfuric acid attack, but unfortunately, the thermal expansion of MAS is many times larger than that of LAS or AS.

The approach taken to reduce the thermal expansion of MAS while retaining its chemical stability is to develop a mixed-phase material which is predominately MAS but contains a second phase material with a lower expansion coefficient. Work conducted by General Electric on MAS with aluminum titanate (MAS-AT) and MAS with LAS indicates that these materials are far superior to LAS in chemical stability (Reference 8-37). A reduction in the expansion coefficient has been realized from this development effort. However, fusion-sintered and nucleated LAS/MAS, the best material to date, has an expansion coefficient several times larger than AS.

Protective coatings applied to the regenerator core material are also being investigated. However, test regenerator cores with coated inserts indicated chemical instability when subjected to state-of-the-art environmental rig tests (Reference 8-31).

8.4.2 Fabrication Methods

Several processes are available for the fabrication of ceramic cellular structures. They include (1) extrusion, (2) calendering, (3) embossing, and (4) coated paper wrapping.

The extrusion process is the simplest of the fabrication methods (Reference 8-31). The ceramic material in the form of a glass frit is mixed with an organic binder. The purpose of the binder is to give the mixture the appropriate handling and forming strength at a high ceramic filler load. The binder system must also be compatible with subsequent ceramic processing steps which include bonding, binder burn-out, and final fixing. The mixture is then forced through a die with the required geometric configuration. Segments are sliced off of a continuously extruded length to the required thickness. The segments are then assembled into a full size regenerator core.

Calendering is a rolling technique used by the plastics industry to fabricate thermoplastic sheets. This process was adopted to produce ribbed sheets suitable for regenerator fabrication (References 8-38 and 8-39). The calendering process, as it pertains to regenerator fabrication, is more correctly described as extrusion-embossing. Thin ribbons are formed by an extrusion process, as earlier described. Ribs are then embossed onto the ribbon as it passes through two calender rollers. The ribbed ribbon is then wound into a disc to the required diameter on a fixture. To insure an adequate bond between the rib tops and the ribbon back, heat or a solvent coat is applied to the ribbon back.

Embossing and calendering differ only with regard to the initial ribbon thickness and the embossing technique. Calendering utilizes two rollers to reduce the ribbon thickness and then the ribbon is embossed while the embossing process utilizes only one roller and an embosser.

The coating paper wrapping process utilizes a low ash paper ribbon to support the ceramic material during the fabrication process (Reference 8-31). The ceramic material, in the form of a slurry, is attached to the paper, which is crimped into the appropriate shape before core winding. Two ribbons are used to produce a sinusoidal geometric configuration.

After the ceramic cellular core structure has been formed, it is fired at a low temperature to remove the organic binder or the low ash paper. This is followed by a higher temperature firing which brings about the glass to glass-ceramic transformation. The glass frit is used to lower the required firing temperature. The glass to glass-ceramic transformation temperature or devitrification temperature is lower than the sintering temperature that would be required to form the ceramic structure if ceramic structure minerals were used. The surfaces of the core are then machined flat and parallel, to the required thickness, and bonded into the drive gear ring.

Before a given fabrication process is employed, a number of engineering factors must be considered. The cellular wall thickness, porosity, and joint bonding are important from a mechanical standpoint. The mechanical strength of the core is reduced with decreasing wall thickness or increased porosity. On the other hand, a small amount of porosity serves to arrest cracks during slow crack growth from matrix thermal shocking. Excessive material at the bond joint will reduce the core strain tolerance.

The geometric configuration to include the core wall thickness must be considered from a thermodynamic performance standpoint. Tests were conducted by Ford (Reference 8-31) on several matrix configurations: (1) rectangular fin, (2) isosceles triangular, and (3) sinusoidal triangular. Results indicate that the rectangular fin is more efficient than the sinusoidal triangular passage of equivalent hydraulic diameter, flow length, and matrix wall material thickness. Test results also indicate that the isosceles triangular configuration is more efficient than the sinusoidal configuration of equivalent hydraulic diameter and flow length in spite of a greater matrix wall thickness. The isosceles triangular configuration appears to be equivalent to the rectangular configuration with a matrix aspect ratio approaching one and with equivalent manufacturing processes. Thinner walls result in increased performance; however, the mechanical strength is decreased.

Of the fabrication techniques, the extrusion method yields the most uniform wall thickness and cellular structure. In addition, this technique is used to manufacture the isosceles triangular matrix configuration which demonstrates desirable thermodynamic characteristics. The drawback associated with the extrusion processes is the difficulty of obtaining thin walls for increasing performance while fabricating large pieces.

The advantage in the coated paper wrapping approach is the wall porosity obtained, which serves as slow crack growth arrester. A wall thickness as low as 0.003 inch, with uniform cells and good joint bonding can be obtained from this technique; however, the sinusoidal triangular cellular structure manufactured from the coated paper wrapping process is not the most desirable geometric configuration from a thermodynamic performance standpoint. In addition, the use of the low ash paper utilized in this process increases the manufacturing cost.

An advantage of the calendering and the embossing processes, which are quite similar, is that they both employ a single ribbed ribbon without a paper carrier to fabricate a rectangular cellular matrix.

Since the thermodynamic performance of the regenerator is dependent upon the cellular wall thickness, fabrication processes which are capable of producing thinner walls are under investigation. A process under development by Owens-Illinois utilizes thin-walled (0.001-inch) glass ceramic tubes to produce a hexagonal geometric cellular configuration with a wall thickness of 0.002 inch. The glass

ceramic tubes are assembled into a stack with closed ends. The assembly is then heated, and the tubes expand due to the internal gas pressure, forming a hexagonal cell structure.

8.4.3 Demonstrated Performance

Regenerator durability tests conducted by Ford (Reference 8-40) indicate that cores fabricated from AS and MAS show promise for meeting the DOE/Ford regenerator program objectives of a regenerator life of 10,000 hours at 1472°F (800°C) with a failure rate no greater than 10 percent. Nine AS cores, with running hours between 800 and 7,000 show no signs of serious chemical attack. A single MAS core shows no signs of attack after 5000 hours; however, the core developed thermal stress cracks in the space between the stress relief slots (Reference 8-34). These cracks were noted after 200 hours of engine testing. The cracks propagated into a region below the regenerator seal shoe inner diameter where the core is in a compressive stress state. Additional crack growth was noted when the core inlet temperature was accidentally increased from 1472°F (800°C) to 1562°F (850°C).

Improvements in design coupled with improved materials have allowed the successful demonstration of regenerators which are capable of operating at temperatures of 1800°F in the engine environment.

8.5 CONCLUSIONS

The adoption of a mass-produced Brayton or Stirling engine for vehicular application is dependent upon the demonstration of engines with improved fuel economy, controlled emissions, and low cost. The high cost and limited operating temperatures of metallic materials necessitate the investigation and advancement of ceramic technology for these heat engines.

In general, demonstration of the feasibility of ceramics in Brayton engines has proceeded with encouraging success. Ceramic heat exchangers which can operate to a temperature of 1800°F (982°C) have demonstrated engineering feasibility along with the capability to be mass produced with current state-of-the-art technology.

Structural ceramic components which can operate to a temperature of 2500°F (1371°C) are near the point of demonstrating engineering feasibility for stationary components. Recent developments in ceramic turbine wheel fabrication are promising. However, the future success of ceramic engine programs will depend upon additional materials and processes development.

Silicon carbide and silicon nitride are the candidate materials for the high stress, high temperature components of advanced

Brayton and Stirling engines. Improved design techniques that utilize ceramic materials to their fullest potential will greatly enhance ceramic component feasibility and reliability.

8.6 RESEARCH AND DEVELOPMENT RECOMMENDATIONS

No fundamental research in ceramic technology is required with respect to the metallic configuration heat engine application. To make the advanced configuration heat engines a reality, research in ceramics technology is required. Although the actual ceramic materials are already identified (invention of a novel material is not implied), the applications to specific components do raise some fundamental questions in four general categories:

- (1) Material Formulation. Determine appropriate compositions and impurity levels for optimum properties in
 - (a) Turbine wheel, nose cone, stators, and combustor applications (SiC; Si₃N₄; others?).
 - (b) Regenerator core application (MAS; AS; Si₃N₄; ceramic composites; others?).
- (2) Ceramic Raw Material Processing. Determine the optimum trade-off between properties achieved at given impurity levels and the cost of refining the abundant raw materials to attain these impurity levels; demonstrate the cost-effectiveness of producing the ceramic material in a pilot plant scalable to mass production quantities.
- (3) Ceramic Structure Fabrication Processes. Demonstrate the production of formed parts, with adequate properties and high reproducibility and yields, from production grade raw materials; develop processes to permit concrescence of simple-shape components into complex monolithic structures, on a mass production basis; demonstrate integrated ceramic design, testing, and nondestructive evaluation techniques suitable for mass production.
- (4) Ceramic/Metallic Structures Joining and Bonding. Develop technique for joining or bonding ceramic materials of the selected compositions to contiguous metallic structures; techniques should emphasize joint strength, sealing (as required) and alleviation of stress concentrations.

NOTE

Fact-finding visits by ATSP team members in connection with the subject matter of this section were made as follows:

Chrysler Corporation, Detroit MI	Nov 1976
	July 1977
Ford Motor Company, Dearborn MI	Nov 1976
	July 1977
General Motors Technical Center, Warren MI	Nov 1976
Detroit Diesel Allison, Indianapolis IN	Dec 1976
Garrett AiResearch Casting Company, Torrance CA	Jan 1977
	July 1977
Garrett AiResearch Manufacturing Company, Phoenix AZ	Feb 1977
Carborundum Corporation, Niagara Falls NY	Mar 1977
Coors Porcelain Company, Golden CO	Mar 1977
General Electric Research and Development Center, Schenectady NY	Mar 1977
G.T.E Sylvania, Inc., Towanda PA	Mar 1977
Norton Company, Worcester MA	Mar 1977
Owens-Illinois Glass Company, Toledo OH	Mar 1977
Williams Research Company, Walled Lake, MI	July 1977
Advanced Materials Engineering, Ltd., Gateshead, England	Sept 1977
Audi NSU Auto Union, Neckarsulm, West Germany	Sept 1977
British Leyland, Coventry, England	Sept 1977
Daimler-Benz AG, Stuttgart, West Germany	Sept 1977
Degussor, Hanau, West Germany	Sept 1977
MAN, Nuremberg, West Germany	Sept 1977
Philips Research Laboratories, Eindhoven, The Netherlands	Sept 1977
Ricardo & Company Engineers, Shoreham-by-Sea, England	Sept 1977
Saab-Scandia, Trollhattan, Sweden	Sept 1977
United Stirling, Malmo, Sweden	Sept 1977
United Turbine, Malmo, Sweden	Sept 1977
Volkswagenwerke, Wolfsburg, West Germany	Sept 1977

SECTION 9

COMPARISONS OF ALTERNATIVE ENGINES FOR AUTOMOTIVE APPLICATIONS

9.1 INTRODUCTION

Detailed information about conventional and advanced automotive engines has been discussed in previous chapters. In this section, this information is summarized, and the potential of advanced alternative heat engine power systems (Brayton and Stirling) is compared with that of the evolving conventional power systems (uniform-charge Otto, diesel, stratified-charge Otto). Factors considered in this comparison include fuel economy, exhaust emissions, multifuel capability, use of advanced materials, and cost/manufacturability.

The need for conservation in the use of petroleum for transportation has led to the passage of Federal fuel economy standards which require an increase from a sales-weighted-average of 18 mpg in 1978 to a sales-weighted-average of 27.5 mpg in 1985 based on the composite driving cycle. The passage of these Federal fuel economy standards has had and will continue to have a significant effect on the type of vehicles produced by the automotive industry. This is leading to the development of lighter weight vehicles and more fuel efficient conventional engines. The impact of these fuel economy standards must be considered when making comparisons of power systems for automotive use beyond the 1985 time period. The baseline vehicles in Table 9-1 were chosen for the comparisons in anticipation of further vehicle downsizing and the use of lighter weight materials to meet the fuel economy standards.

Table 9-1. Baseline Vehicle Characteristics

Vehicle Size	Horsepower	Curb Weight, lb	HP/Weight
Small	60	1750	0.0343
Full sized	120	3200	0.0375

Since the time period covered in this study extends to the end of the century, each power system is evaluated for its ability to meet not only the current Federal emissions standards (0.4 g/mi HC, 3.4 g/mi CO, 1.0 g/mi NO_x) but also possible future emissions requirements (0.4 g/mi HC, 3.4 g/mi CO, 0.4 g/mi NO_x). Consideration is also given to presently unregulated emissions, such as particulates, noise, and odor.

The following power systems are included in the comparisons in this section:

- (1) Conventional Engines
 - (a) Uniform-charge Otto (Reciprocating)
 - Naturally aspirated
 - Turbocharged
 - (b) Diesel
 - Naturally aspirated
 - Turbocharged
- (2) Advanced Conventional Engines
 - (a) Uniform-charge Otto (Rotary)
 - (b) Stratified-charge Otto
 - Reciprocating
 - Rotary
- (3) Advanced Alternative Engines
 - (a) Brayton (metal and ceramic)
 - Free turbine
 - Single shaft
 - (b) Stirling (metal and ceramic)

In making comparisons of the various alternative power systems, it is important that these comparisons be made at the vehicle level and that a consistent methodology be followed. The method used in this study is discussed in the next section. Other parts of this section cover the following comparison areas:

- (1) Engine weight and size.
- (2) Engine torque-speed characteristics
- (3) Transmission requirements.

- (4) Engine fuel consumption and vehicle fuel economy.
- (5) Vehicle emissions.
- (6) Fuel requirements.
- (7) Material requirements.
- (8) Engine cost and manufacturability.
- (9) Development status and projected availability.

The uncertainty in the data/information used in making the comparisons varies considerably for the various engines. In the case of conventional engines, most of the data/information is for production engines or engines near production status. For advanced conventional engines, the data/information is for limited-production engines or prototype engines which have undergone considerable testing. Uncertainties in the characteristics of advanced alternative engines, especially those using ceramic components, are much greater than for conventional and advanced conventional engines. These differences in development status should be kept in mind when making comparisons of the various power systems.

9.2 COMPARISON METHODOLOGY

Central to the comparison of vehicles with alternative power systems is the concept of an Otto-engine-equivalent (OEE) vehicle. This concept originated during the earlier APSES study, Reference 9-1, at JPL. To the individual customer, the OEE vehicle should be indistinguishable in transportation function and driving behavior from the baseline vehicle. In comparison with the baseline vehicle, this concept requires the OEE vehicle to have the same passenger and luggage space, same accessories, same drag coefficient and frontal area, same operating range, and equivalent performance.

The meaning of equivalent performance requires further comment. Each alternative engine is sized to provide the same vehicle performance as the baseline system according to some performance criteria. The following four performance criteria have been evaluated during this study: 0-60 mph acceleration time, 10-second acceleration distance, 40-60 mph acceleration time, and a combination of the preceding three criteria. The combined criteria were satisfied for an alternative power system by minimizing the RMS deviation from baseline performance for the three criteria.

Vehicle performance is calculated using a vehicle computer simulation program. Appropriate power system weights and engine torque-speed characteristics are used for each alternative power system. Vehicle weight propagation effects are included in the calculation procedure to properly account for the influence of power system weight on vehicle design. This procedure assumes that each alternative vehicle is designed with the same degree of optimization. This procedure yields the appropriate engine horsepower and vehicle weight for

each power system. A detailed discussion of this method for the horsepower sizing of alternative power systems is given in Reference 9-2.

Once the engine horsepower and vehicle weight for each OEE vehicle has been determined, fuel economy is calculated using a vehicle computer simulation program. This program uses steady-state engine map data to predict vehicle fuel economy for the city, highway, and composite driving cycles. The computer program has been calibrated for a 1978 Pontiac Sunbird (3000 lb inertia weight) with a four-cylinder, 151 CID engine and a four-speed manual transmission. The results of this calibration are given in Table 9-2.

The fuel consumption map and idle fuel flow for this calculation were based on test data from Bartlesville Energy Research Center (BERC). The predicted fuel economy values are in reasonable agreement with the EPA values for the city, highway, and composite driving cycles. Since the calculated fuel economies are used only in making comparisons, the accuracy demonstrated by this calibration is considered adequate. The computer program does not include engine transient effects (response time, etc.), which could be important factors for the advanced alternative power systems.

9.3 ENGINE WEIGHT AND SIZE

Weight information for each alternative power system is needed to support the OEE vehicle concept described in the previous section. Data on engine weights were obtained from technical discussions with domestic and foreign automobile manufacturers and government laboratories, as well as technical publications. Representative weight characteristic curves were established for each engine type using the data base available.

The data bases which were used to establish the weight characteristics for conventional engines (uniform-charge (UC) Otto, naturally-aspirated (NA) diesel, turbocharged (TC) diesel) are given in Figures 9-1, 9-2, and 9-3. For conventional engines of a given type,

Table 9-2. Computer Program Calibration for 1978 Pontiac Sunbird

Driving Cycle	Fuel Economy, mpg	
	Predicted	EPA Measurement
City	20.5	22
Highway	34.1	32
Combined	25.0	26

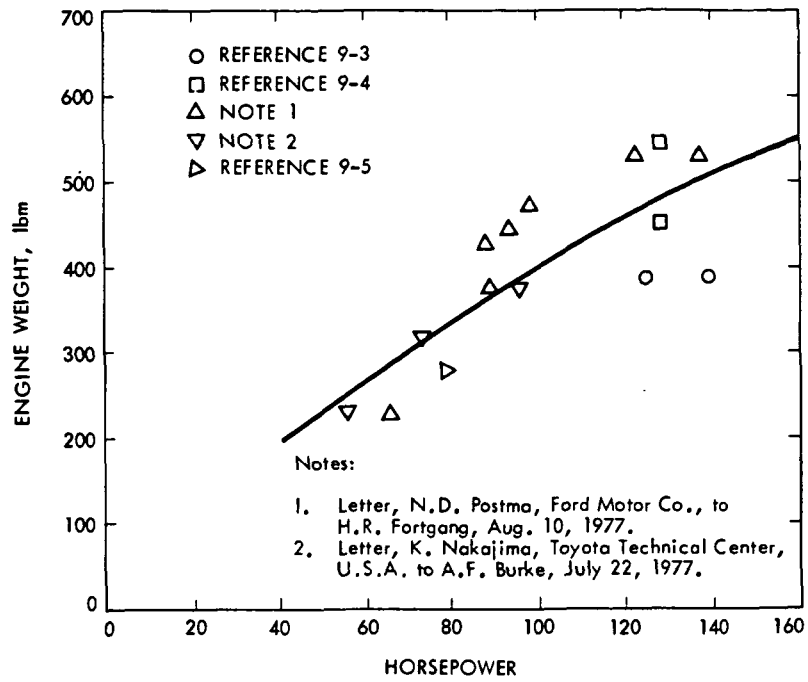


Figure 9-1. Engine Weights for Uniform-Charge Otto

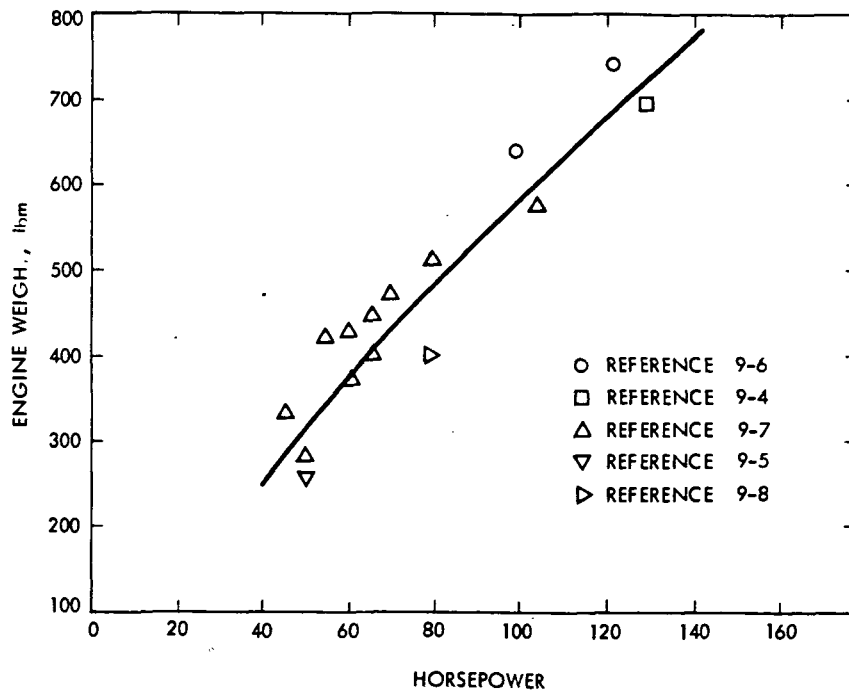


Figure 9-2. Engine Weights for Naturally-Aspirated Diesel

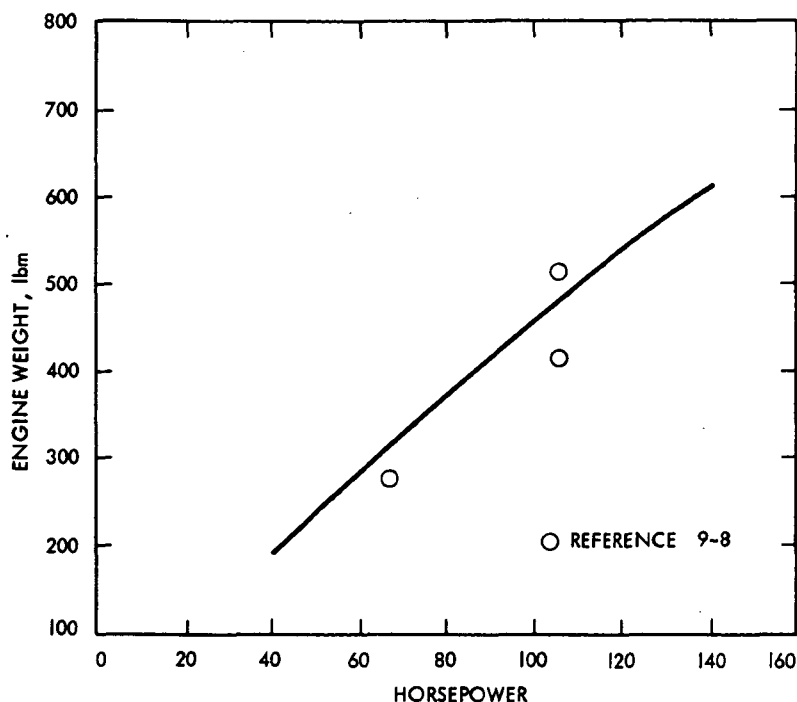


Figure 9-3. Engine Weights for Turbocharged Diesel

there are considerable differences in engine weights, as is shown by the data bands. The lighter weight engines are representative of the more recent designs. For example, the data points from Reference 9-3 in the UC Otto plot represent the Peugeot-Renault-Volvo V-6 engine. The older engine designs generally fall on the upper edge of the data band. The solid lines through the data were used in the OEE vehicle calculations for these engine types.

The weight characteristics for advanced conventional engines (rotary UC Otto, reciprocating stratified-charge (SC) Otto, rotary SC Otto) are given in Figures 9-4, 9-5, and 9-6. In this case, much less data is available for establishing the characteristic because of the limited number of engines of this type which have been built. Again, the solid lines were used in the OEE vehicle calculations.

The weight characteristics for advanced alternative engines (free-turbine (FT) Brayton, single-shaft (SS) Brayton, Stirling) are shown in Figures 9-7, 9-8, and 9-9. Using current weight technology, the data in the plots represent some current developmental alternative engines. As development of these engines continues, it is expected that engine weights will approach the characteristics which have been assumed for the OEE vehicle calculations. In the OEE vehicle concept, engine weight plays an important role in determining projected fuel economy. Thus, it is very important that future weight technology permit the advanced alternative engines to achieve the assumed weight characteristic. The effect of engine weight on vehicle fuel economy will be discussed in a later section.

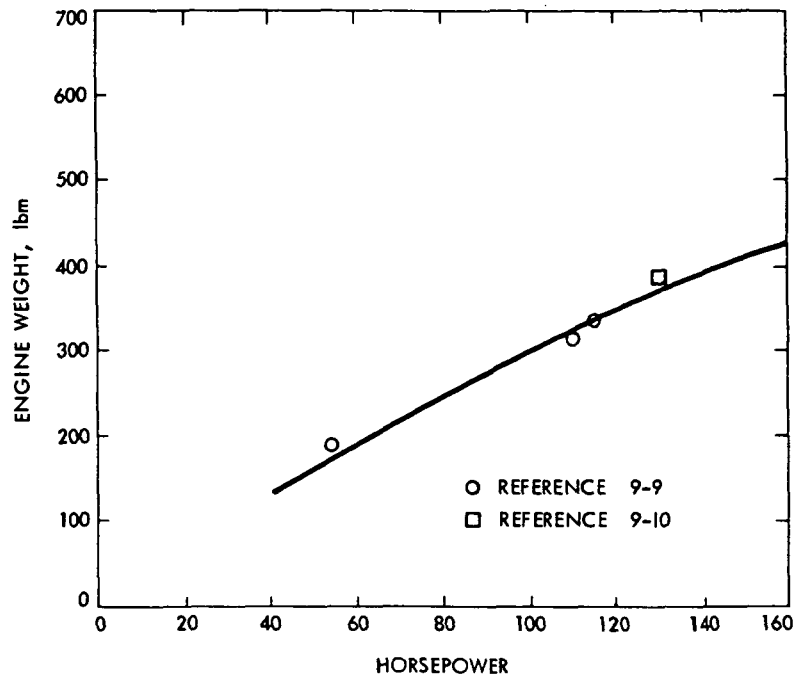


Figure 9-4. Engine Weights for Uniform-Charge Otto (Rotary)

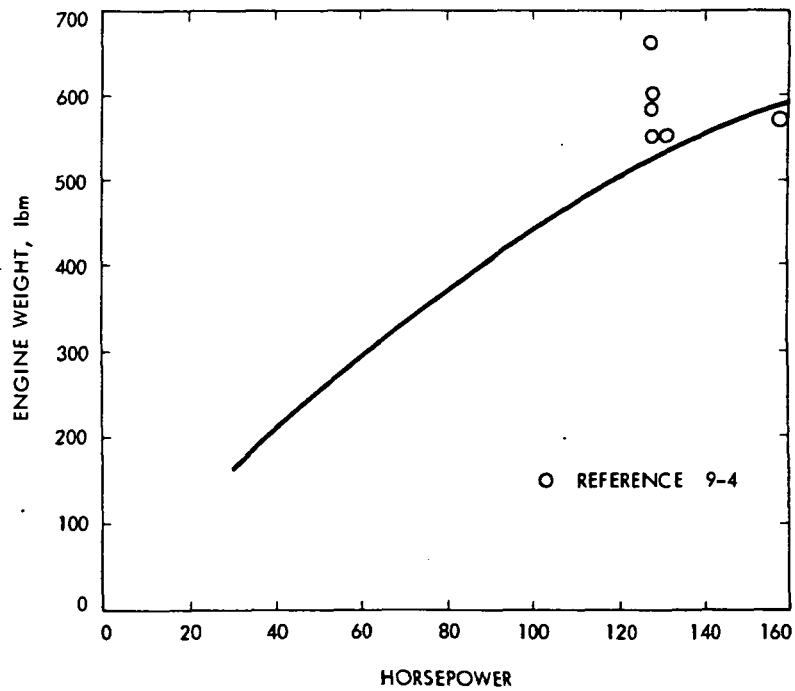


Figure 9-5. Engine Weights for Stratified-Charge Otto

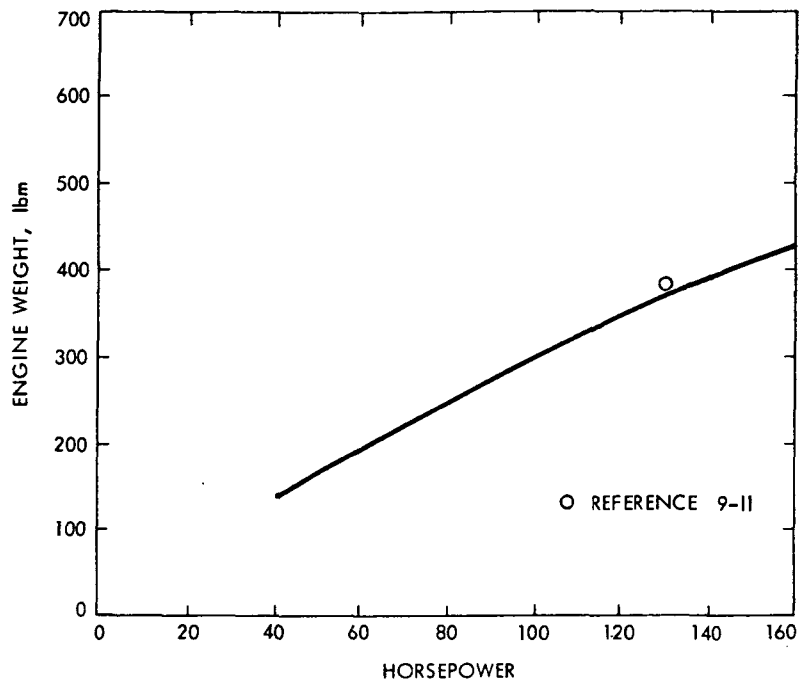


Figure 9-6. Engine Weights for Stratified-Charge Otto (Rotary)

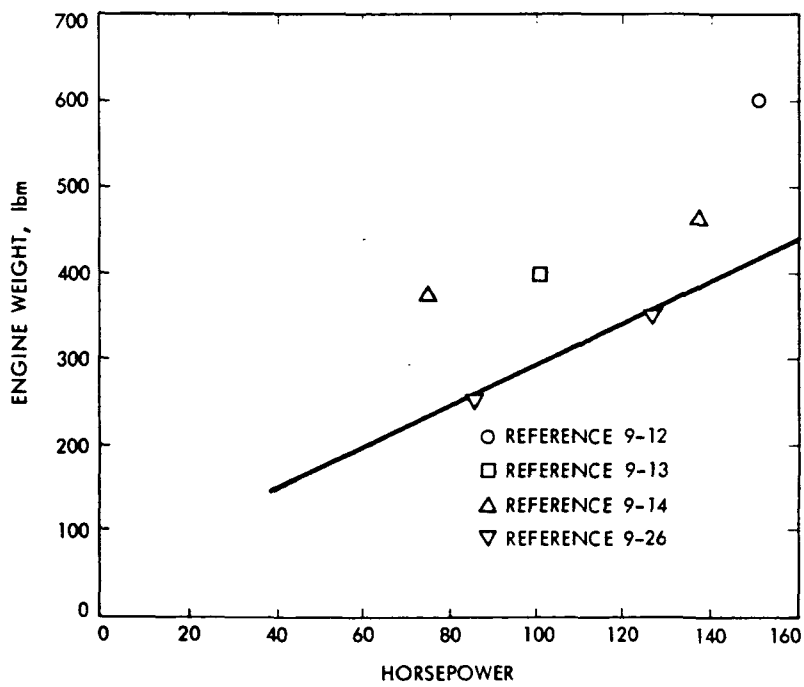


Figure 9-7. Engine Weights for Free-Turbine Brayton

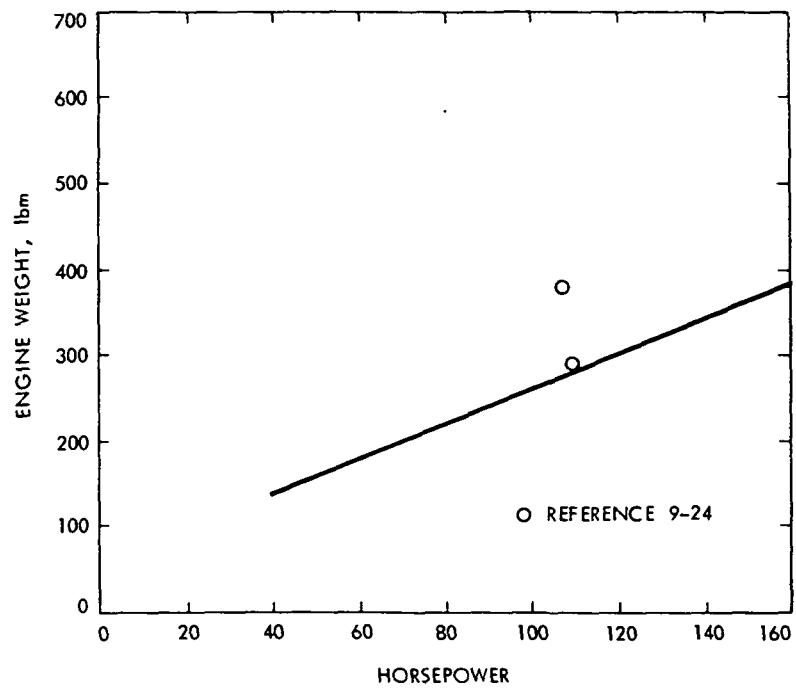


Figure 9-8. Engine Weights for Single-Shaft Brayton

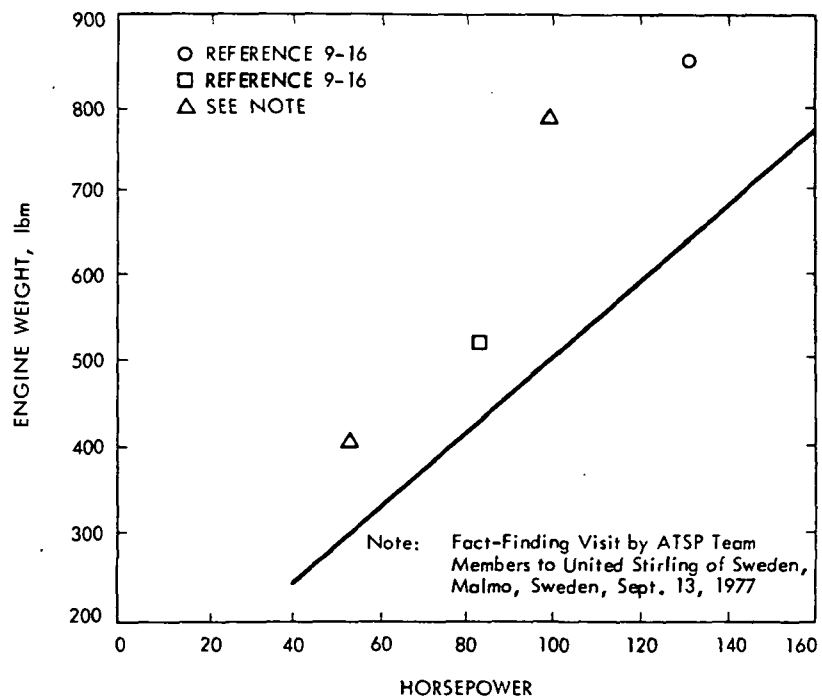


Figure 9-9. Engine Weights for Stirling

Weights for the transmission, battery, and cooling system are given in Figures 9-10, 9-11, and 9-12, respectively. Note that a larger weight battery is required for the diesel engine compared with the other engine types. A significantly heavier cooling system is required by the Stirling engine since it rejects a larger fraction of the total fuel energy to the cooling water. On the other hand, the Brayton engine does not require a radiator.

Total power system weights for each engine type are given as a function of horsepower in Figure 9-13. The total power system includes the engine, all auxiliaries, transmission, battery, and cooling system. The curves for conventional engines and advanced conventional engines are representative of current weight technology. The curves for the advanced alternative engines represent projections at a time when those engines have been developed for passenger car use. These curves show the Stirling power system weight to be about equal to that of the TC diesel. The FT Brayton, SS Brayton, rotary UC Otto and rotary SC Otto power systems show significant weight advantages over other engines considered in this study. The NA diesel power system is considerably heavier than other engines considered. The importance of these weight differences will be discussed later.

The size characteristics of various engines are given in Table 9-3. Most of the data are for engines in the 100-150 horsepower range since size data for smaller engines is limited. This data indicates that the TC Otto and rotary engines require less volume than the other engines of the same horsepower. Most current designs of Brayton and Stirling engines are relatively bulky and do not offer any packaging advantages over the conventional engines.

9.4 ENGINE TORQUE-SPEED CHARACTERISTICS

The shape of the engine torque-speed characteristic is important in determining the acceleration capability of a vehicle. Torque-speed curves, which are representative of each engine type, are needed to establish the horsepower and weight of OEE vehicles with equivalent performance. Using the data base available from automobile manufacturers, government laboratories, and technical publications, characteristic torque curves were developed for each engine type.

The data bases for the torque-speed characteristics for conventional engines (UC Otto, NA diesel, TC diesel) are given in Figures 9-14, 9-15, and 9-16. The engine torque and speed are normalized with respect to the torque and speed at maximum power. The curves through the data bands represent the characteristic which was used in the OEE vehicle calculations. Torque-speed characteristics for the advanced conventional engines (rotary UC Otto, reciprocating SC Otto, rotary SC Otto) are shown in Figures 9-17, 9-18, and 9-19. In this case, the curves are based on rather limited data. The basis for the torque-speed characteristics for the advanced alternative engines (FT Brayton, SS Brayton, Stirling) are given in Figures 9-20, 9-21, and 9-22.

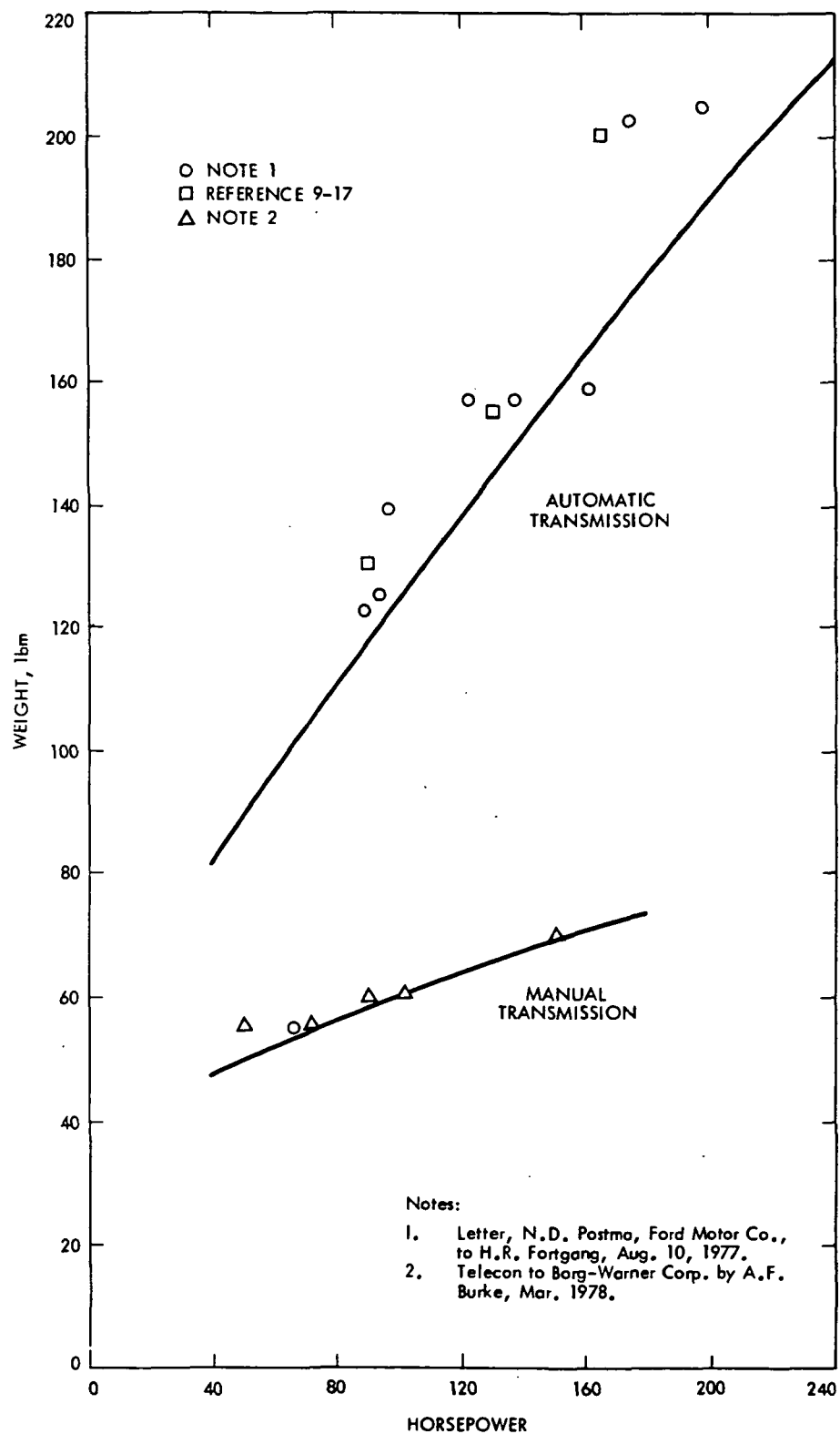


Figure 9-10. Transmission Weights

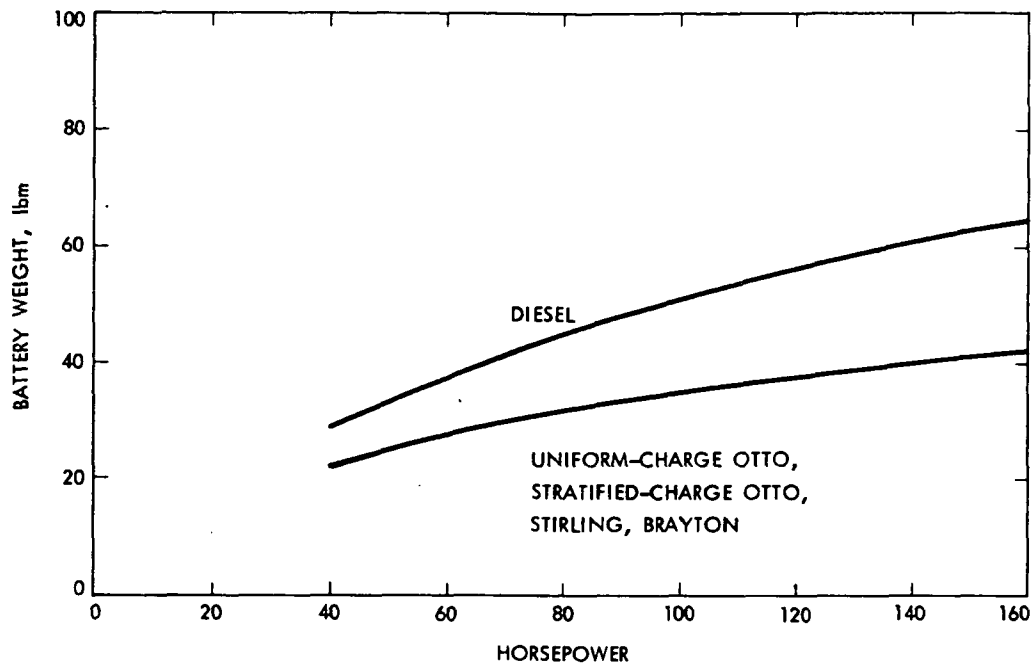


Figure 9-11. Battery Weights

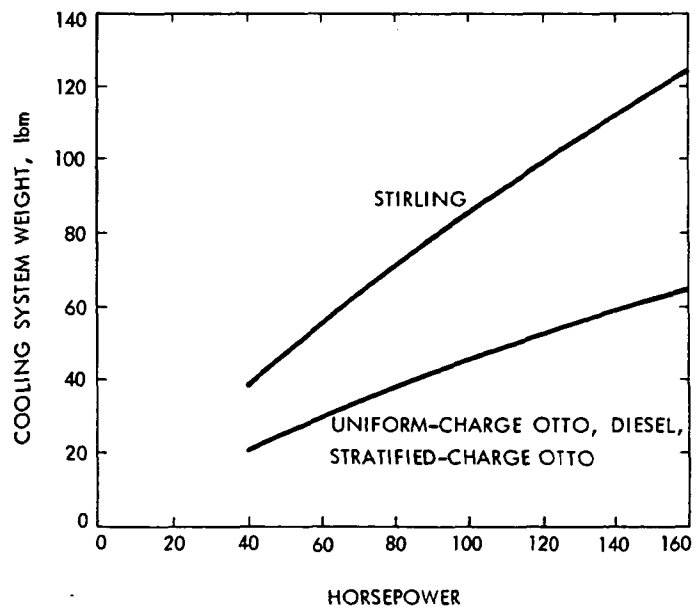


Figure 9-12. Cooling System Weights

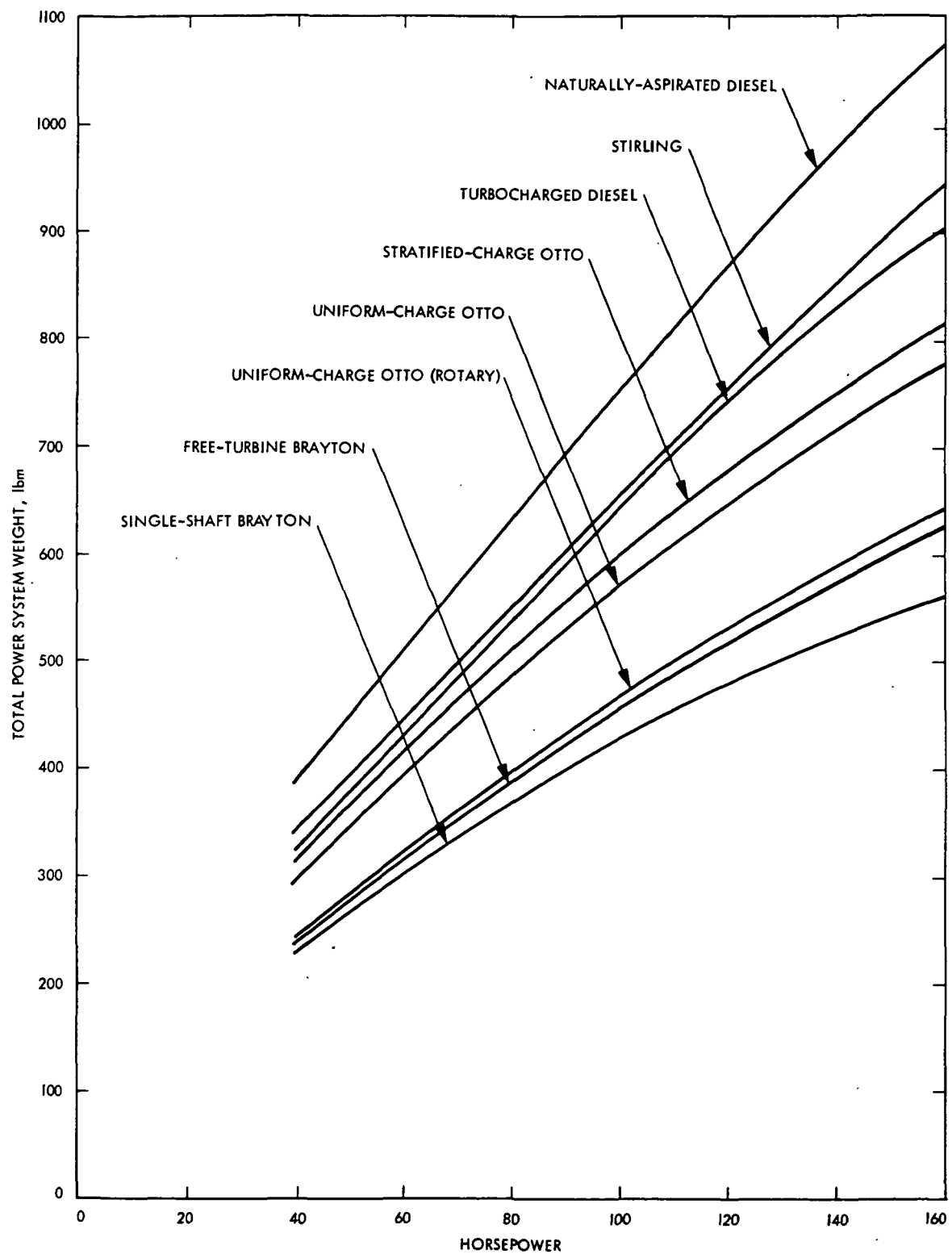


Figure 9-13. Total Power System Weights

Table 9-3. Typical Engine Size Characteristics

Engine Type	Horsepower	Engine Size, in.			
		L	W	H	
Conventional Engines					
UC Otto	130	19	25	25	
NA diesel	120	32	28	30	
	50	17	17	22	
TC diesel	105	31	19	31	
	70	17	17	22	
Advanced Conventional Engines					
Rotary UC Otto	170	18	29	23	
Reciprocating SC Otto	128	30	24	24	
Rotary SC Otto	128	20	27	26	
Advanced Alternative Engines					
FT Brayton (metal)	110	33	33	27	
Stirling (metal)	170	37	26	26	
	84	26	21	23	

In this case, the band is based on actual engine data and on projections of engine performance. This is necessary because of the developmental status of the engines.

A comparison of the torque-speed characteristics for some of the engines is given in Figure 9-23. For passenger car applications, engines with a high torque ratio at relatively low engine speed are advantageous since they exhibit better acceleration performance. These torque curves indicate that both Brayton and Stirling engines have better torque characteristics than those for conventional engines. When using the OEE vehicle concept, these characteristics require the Brayton and Stirling engines to have less horsepower than that of conventional engines for equivalent vehicle performance.

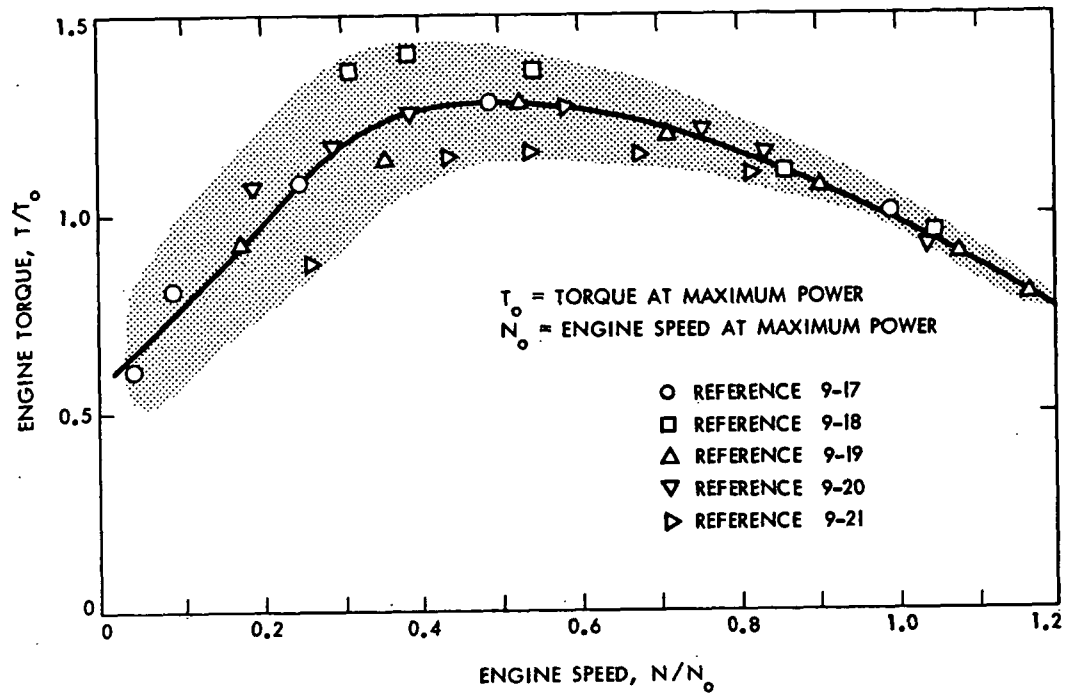


Figure 9-14. Torque-Speed Characteristics for Uniform-Charge Otto

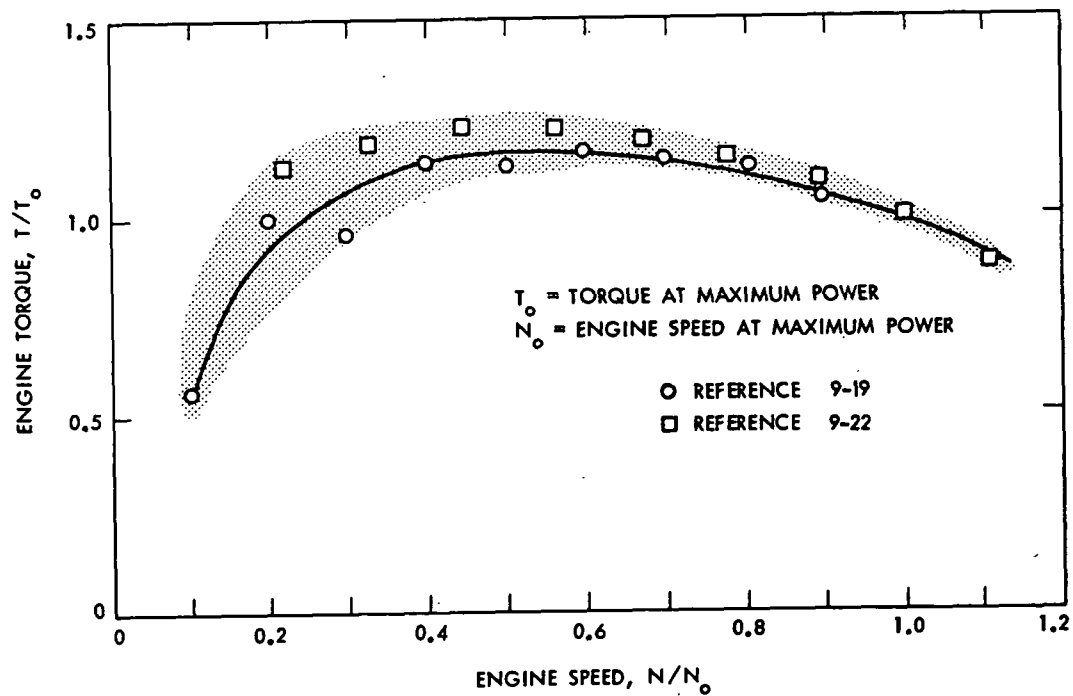


Figure 9-15. Torque-Speed Characteristics for Naturally-Aspirated Diesel

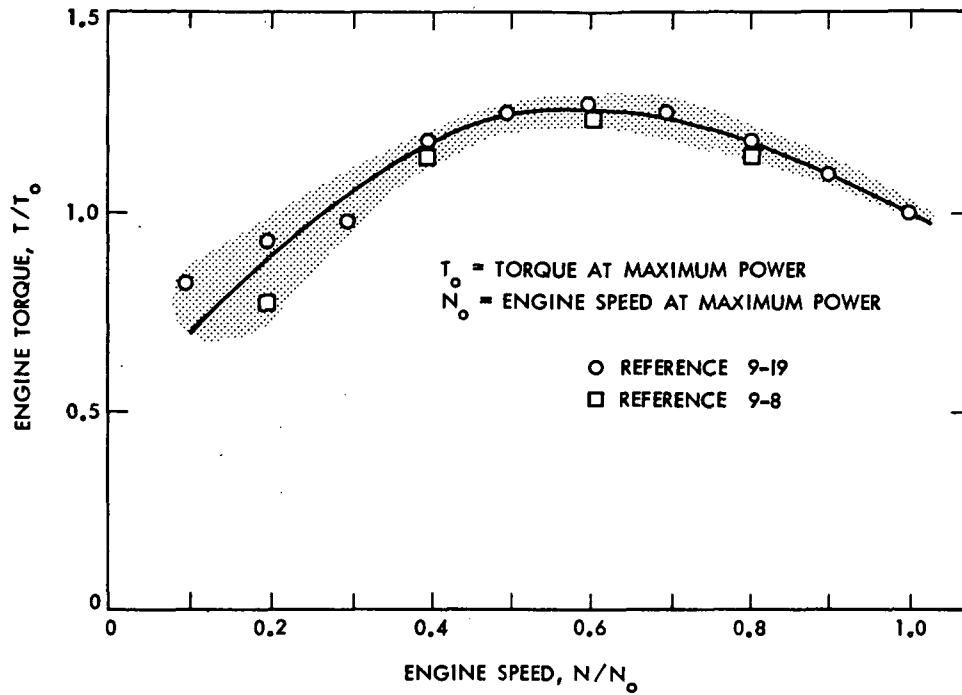


Figure 9-16. Torque-Speed Characteristics for Turbocharged Diesel

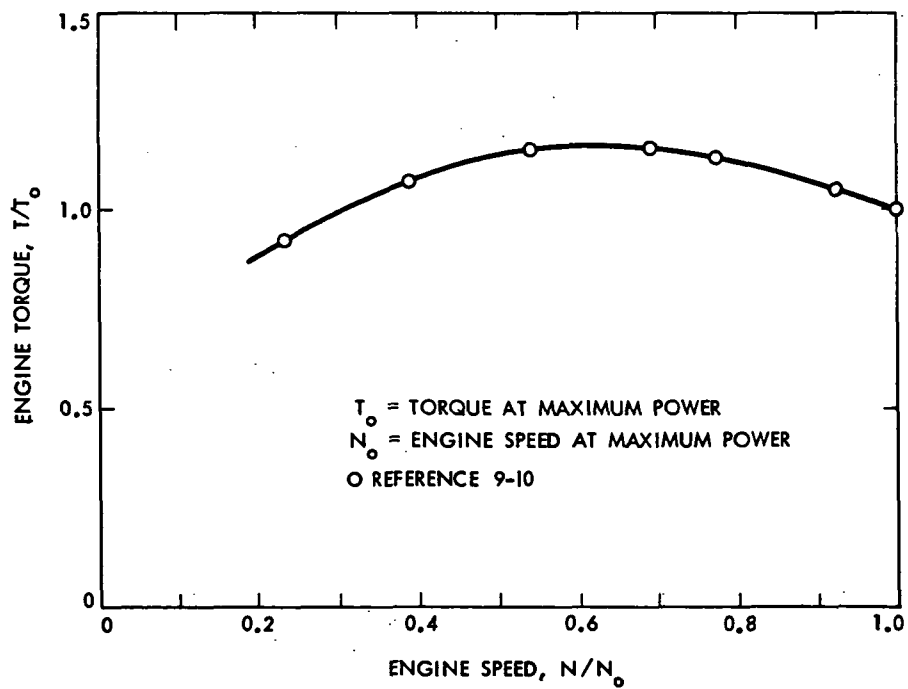


Figure 9-17. Torque-Speed Characteristics for Uniform-Charge Otto (Rotary)

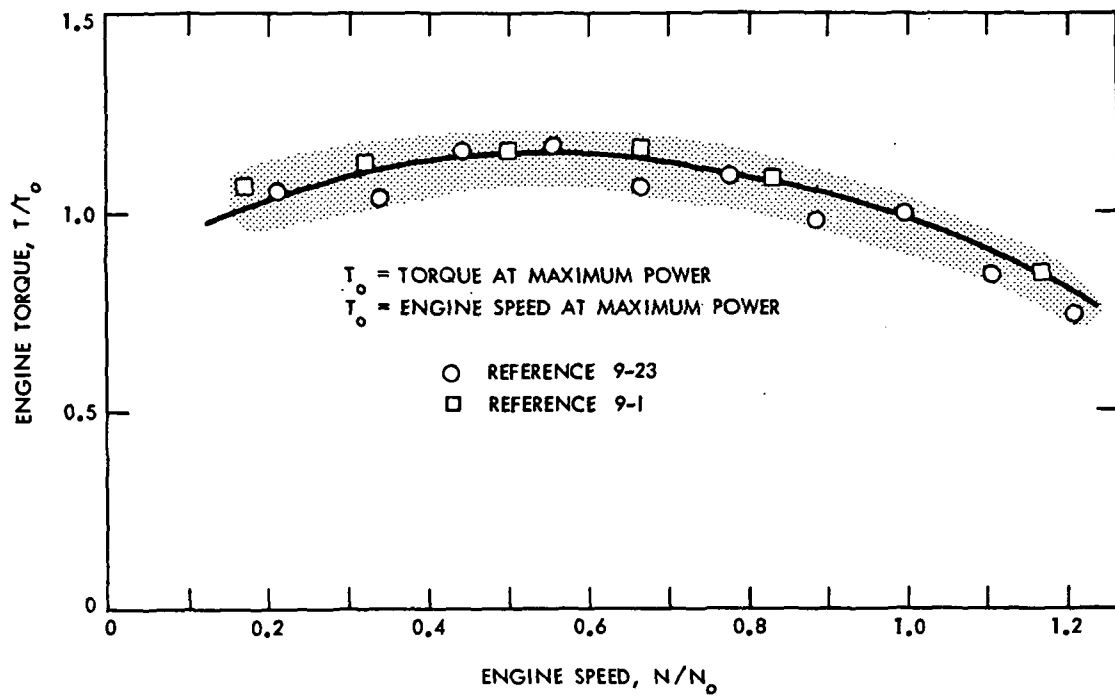


Figure 9-18. Torque-Speed Characteristics for Stratified-Charge Otto

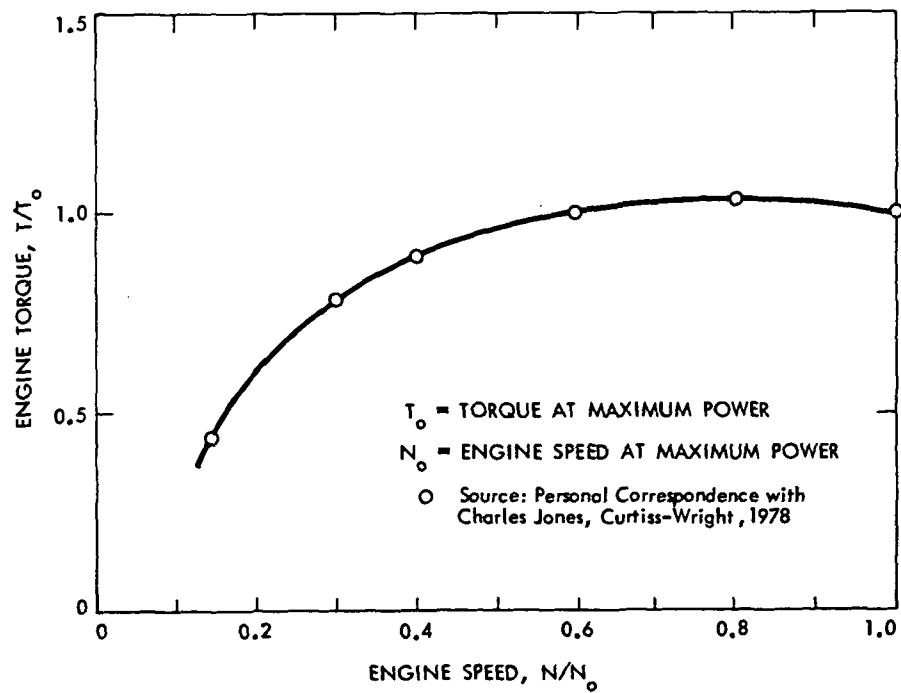


Figure 9-19. Torque-Speed Characteristics for Stratified-Charge Otto (Rotary)

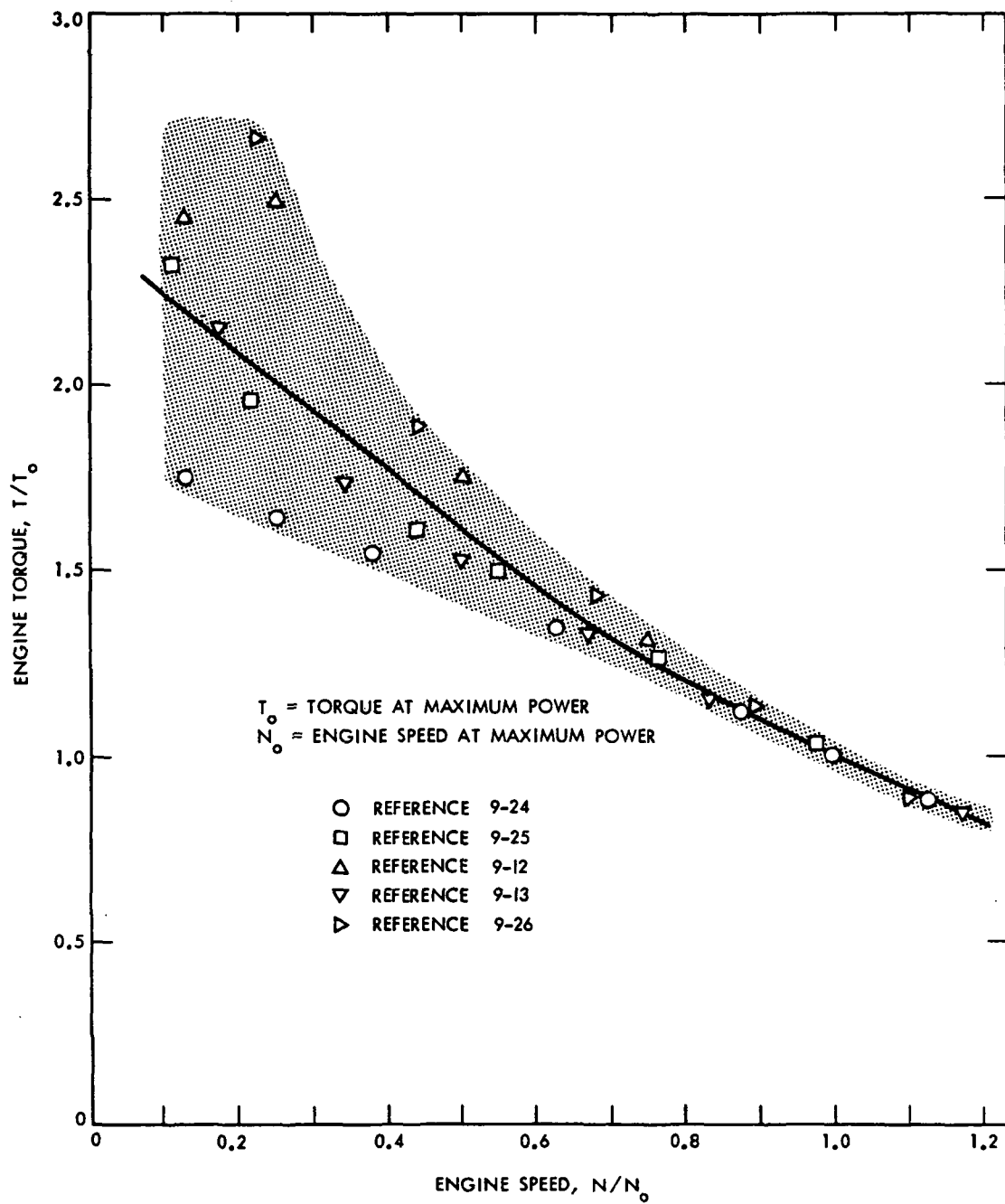


Figure 9-20. Torque-Speed Characteristics for Free-Turbine Brayton

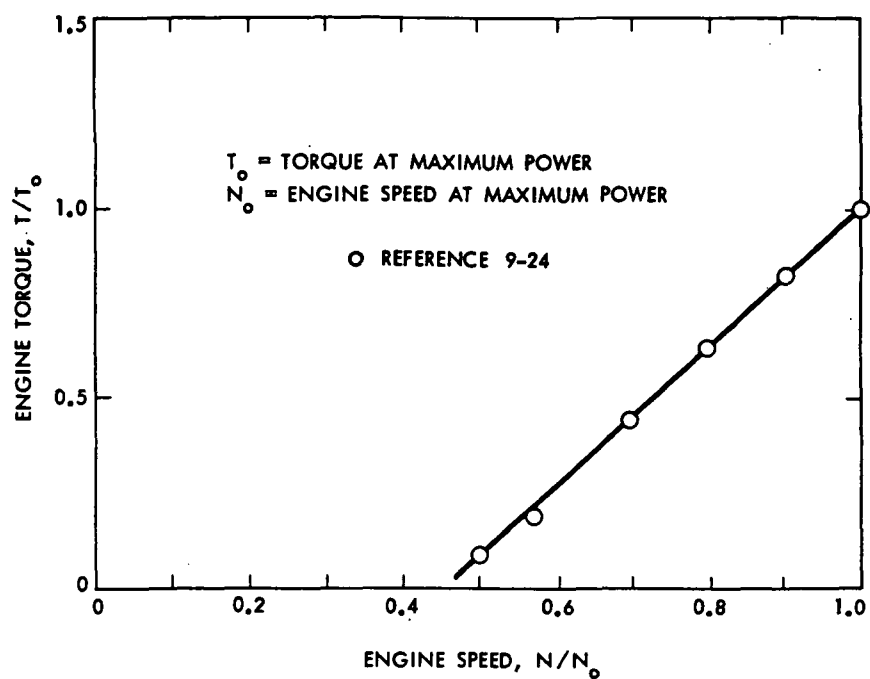


Figure 9-21. Torque-Speed Characteristics for Single-Shaft Brayton

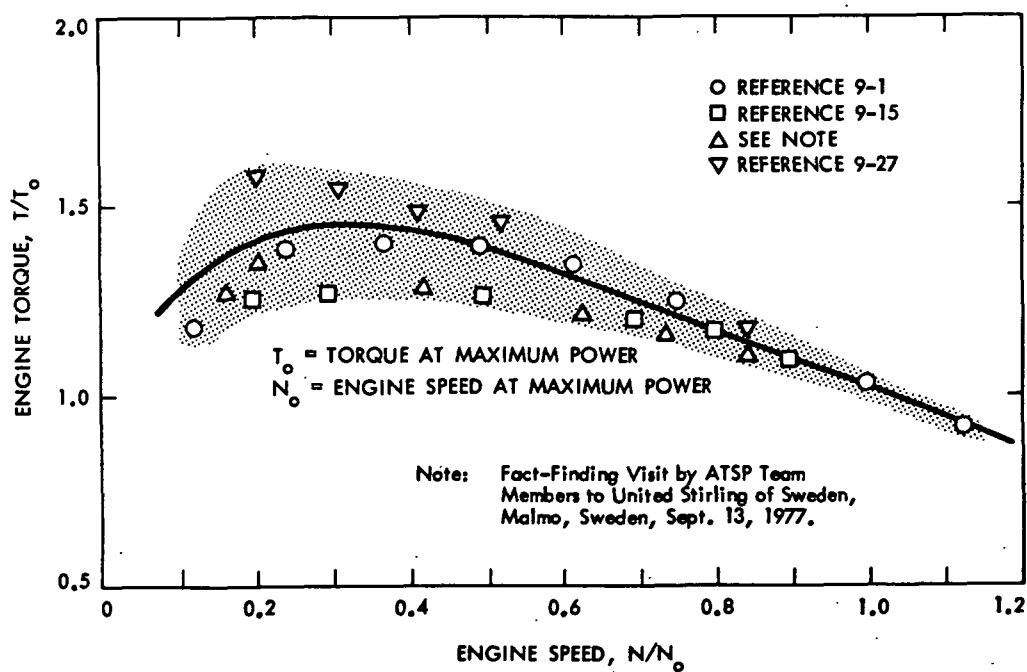


Figure 9-22. Torque-Speed Characteristics for Stirling

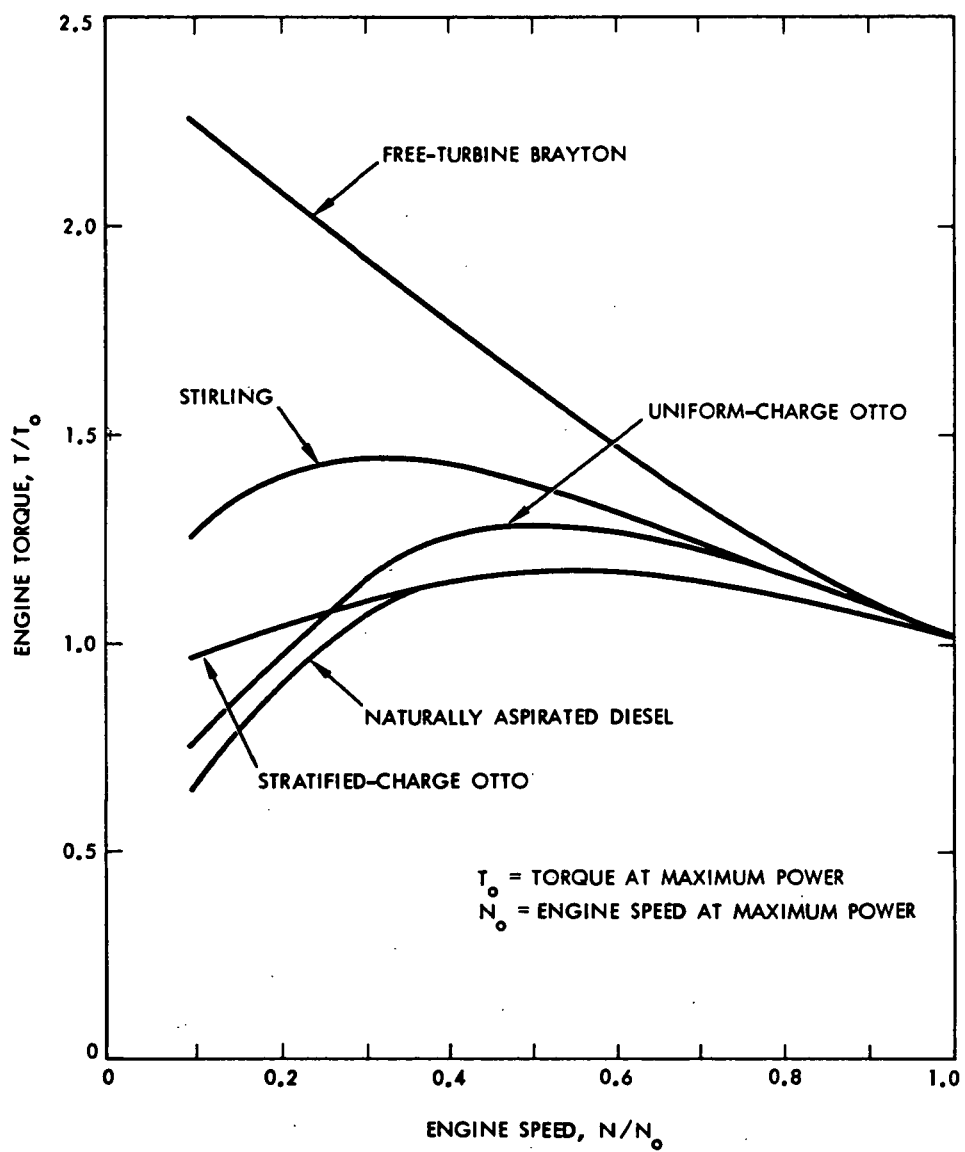


Figure 9-23. Typical Torque-Speed Characteristics

Except for the SS Brayton engine, which requires the use of a continuously variable transmission (CVT), all the engines being compared can utilize the conventional manual and automatic transmissions currently available. Of course, the gear ratios, torque converter characteristics, and shift logic would need to be optimized for each of the engine types. Transmission improvements such as overdrive and torque converter lockup are applicable to the advanced conventional engines and Stirling engines, and would benefit those engines in essentially the same way that they benefit conventional engines. A summary of projected fuel economy improvements using various advanced automatic transmissions in vehicles with conventional engines is given in Table 9-4.

The transmission situation for Brayton engines is more complex than for the other engine types. Brayton engines operate at much higher RPM (50,000-100,000) and thus require a speed-reduction between the turbine and vehicle drive shafts. In addition, their effective inertia is higher and thus drive shaft speed changes of the same magnitude require more power for the Brayton than for engines operating at lower RPM. Hence, it is advantageous to limit Brayton engine speed changes during both accelerations and decelerations as much as possible. For the multishaft FT Brayton, there is no need for a torque converter since the free turbine can perform that function. This can lead to a reduction in driveline weight and packaging volume and improved efficiency. The advantages of combining the engine and transmission in a multishaft Brayton engine are discussed in Reference 9-26. The computer simulations of two-shaft gas turbine-powered cars, which are discussed in this report, were made using an M4 transmission.

As noted previously, the SS Brayton requires the use of a CVT to match engine and vehicle drive shaft speeds. Such transmissions are not in current use on production vehicles, but a prototype model of a CVT using the hydromechanical, power-split approach is being developed under DOE contract (Reference 9-28). This transmission is currently being tested in a Chevrolet Nova equipped with a conventional six-cylinder engine. Development of a durable, high efficiency CVT is clearly critical to the introduction of SS Brayton engines. The CVT can, of course, be used with the other types of engines and would also improve their performance and fuel economy.

9.6

ENGINE FUEL CONSUMPTION AND VEHICLE FUEL ECONOMY

9.6.1

Engine Fuel Consumption Characteristics

Engine fuel consumption characteristics for each engine type are required before vehicle fuel economy projections can be made using the computer simulation program. Data on engine fuel consumption were obtained during technical discussions with domestic and foreign automobile manufacturers and Government laboratories, as well as

Table 9-4. Summary of Projected Fuel Economy Improvements Using Various Advanced Automatic Transmissions

Type	Transmission				Fuel Economy Improvement, %		
	No. Gears	Overall Gear Ratio Range	Lockup	Efficiency, %			
					Urban	Highway	Composite
Automatic	3	3.0 to 1	No	-	0(base)	0	0
Automatic	3	3.0 to 1	Yes	-	5-10	5-10	5-10
Automatic	4	4.0 to 1	No	-	3- 5	15-20	8-11
Automatic	4	4.0 to 1	Yes	-	10-15	20-25	14-19
Continuously Variable	-	7.0 to 1	-	80-90	15-20	20-25	17-22
Continuously Variable	-	7.0 to 1	-	85-90	18-22	25-30	21-25

from technical publications. Representative fuel consumption maps were established for each engine type using the data base available.

The fuel consumption maps which were used to represent conventional engines (UC Otto, NA diesel, TC diesel) are given in Tables 9-5, 9-6, and 9-7. Values of brake specific fuel consumption (bsfc) are given as a function of speed fraction and power fraction. Engine speed is normalized with respect to the maximum engine RPM. Engine power is normalized with respect to the maximum power attainable at each RPM. Representative maps for the advanced conventional engines (rotary UC Otto, SC Otto, rotary SC Otto) are given in Tables 9-8, 9-9, and 9-10. Similar bsfc information for the advanced alternative engines (FT Brayton, SS Brayton, Stirling) is given in Tables 9-11 through 9-14. The Brayton fuel consumption characteristics are for engines with 100 hp or more. Size effects for Brayton engines of lower horsepower will be discussed later.

It is of considerable interest to compare the fuel consumption characteristics of the various engines. This is done in Figures 9-24, 9-25, and 9-26 where brake specific fuel consumption (bsfc) is given as a function of power fraction at the midpoint of the operating RPM range for each engine. The power is normalized with respect to the maximum power attainable at the midpoint of the operating RPM range. These data are taken from the fuel consumption maps in Tables 9-5 through 9-14. As indicated in Figures 9-24 and 9-25, there are significant differences between the bsfc of the various conventional engines. These differences are apparent in both the minimum bsfc values and the relatively steep rise in bsfc at part-load conditions. The bsfc values of Stirling and Brayton engines are shown in Figure 9-26. The differences between bsfc curves for engines using metal and ceramic components are readily apparent. For the Brayton engines, the bsfc values given are for a 100-120 hp engine. The influence of size effects on the bsfc of smaller Brayton engines is of considerable importance because many of the downsized passenger cars of the 1985-90 period will require engines smaller than 100 hp. Based on the Brayton engine size-effects analysis discussed in Section 5, the estimated effect of reduced peak horsepower on the bsfc is shown in Figure 9-27 for a range of horsepower and turbine inlet temperatures. The results indicate that (1) size effects degrade the bsfc by 10-15 percent when engine horsepower is reduced from 100 to 50 and (2) that the use of ceramic components, which permit an increase in turbine inlet temperature from 1900 to 2500°F improves bsfc by 10-15 percent. These curves were used in making projections for Brayton engines to account for the effects of size and turbine inlet temperature. In making projections for Stirling engines, it was assumed that the bsfc characteristics would not be degraded by size effects. The method used to account for higher operating temperatures in Stirling engines was based on the analysis of United Stirling in Reference 9-27.

Comparisons of the bsfc of Brayton and Stirling engines with those of conventional engines are shown in Figures 9-28 to 9-30. The Stirling engine with a metal heater-head is compared with other gasoline engines in Figure 9-28. The potential advantage of the Stirling engine is clear if the predicted fuel consumption characteristics of the

Table 9-5. Fuel Consumption Map for the UC Otto Engine (Reference 9-29)

f_{RPM}	f_{HP}						
	0.10	0.25	0.40	0.60	0.75	0.90	1.00
0.227	1.40	0.666	0.568	0.467	0.462	0.465	0.517
0.341	1.38	0.692	0.548	0.458	0.462	0.479	0.540
0.455	1.26	0.667	0.513	0.448	0.540	0.449	0.487
0.568	1.19	0.674	0.537	0.460	0.446	0.439	0.483
0.727	1.32	0.681	0.544	0.470	0.463	0.440	0.490
0.864	1.51	0.700	0.550	0.494	0.483	0.467	0.495
1.000	1.30	0.773	0.592	0.509	0.507	0.478	0.527

$$f_{HP} = HP/HP_{\max@rpm}, f_{RPM} = RPM/RPM_{\max}$$

Table 9-6. Fuel Consumption Map for NA Diesel Engine (Reference 9-19)

f_{RPM}	f_{HP}						
	0.05	0.10	0.25	0.40	0.60	0.75	1.00
0.2	1.46	1.30	0.87	0.61	0.49	0.46	0.45
0.3	1.39	1.21	0.67	0.52	0.46	0.45	0.46
0.4	1.39	1.21	0.67	0.52	0.46	0.45	0.46
0.5	1.48	1.29	0.69	0.53	0.44	0.42	0.58
0.6	1.56	1.35	0.72	0.54	0.46	0.45	0.52
0.8	1.85	1.50	0.81	0.60	0.51	0.49	0.49
1.0	2.19	1.93	1.15	0.81	0.61	0.57	0.57

$$f_{HP} = HP/HP_{\max@rpm}, f_{RPM} = RPM/RPM_{\max}$$

Table 9-7. Fuel Consumption Map for TC Diesel Engine (Reference 9-19)

f_{RPM}	f_{HP}						
	0.05	0.10	0.25	0.40	0.60	0.75	1.00
0.20	1.760	1.390	0.860	0.610	0.510	0.495	0.495
0.30	1.450	1.200	0.710	0.530	0.460	0.440	0.500
0.40	1.560	1.150	0.610	0.500	0.450	0.460	0.500
0.50	1.650	1.150	0.610	0.460	0.440	0.440	0.500
0.60	1.900	1.220	0.590	0.460	0.430	0.430	0.500
0.70	2.140	1.730	0.630	0.460	0.430	0.430	0.500
0.80	2.470	1.400	0.710	0.530	0.440	0.460	0.540
1.00	2.400	1.730	1.090	0.760	0.590	0.540	0.580

$f_{HP} = HP/HP_{max@rpm}$, $f_{RPM} = RPM/RPM_{max}$

Table 9-8. Fuel Consumption Map for Rotary UC Otto Engine (Reference 9-10)

f_{RPM}	f_{HP}						
	0.10	0.15	0.25	0.35	0.60	0.75	1.00
0.231	1.000	0.880	0.750	0.640	0.520	0.497	0.484
0.308	1.000	0.860	0.708	0.594	0.495	0.473	0.473
0.462	1.000	0.860	0.645	0.550	0.466	0.452	0.462
0.615	1.040	0.847	0.650	0.550	0.473	0.451	0.462
0.769	1.210	1.000	0.660	0.605	0.506	0.484	0.484
0.923	1.400	1.200	0.935	0.649	0.541	0.524	0.515
1.000	1.700	1.400	1.100	0.910	0.590	0.550	0.530

$f_{HP} = HP/HP_{max@rpm}$, $f_{RPM} = RPM/RPM_{max}$

Table 9-9. Fuel Consumption Map for SC Otto Engine (Reference 9-1)

f_{RPM}	f_{HP}						
	0.10	0.20	0.30	0.40	0.60	0.80	1.00
0.167	1.02	0.80	0.66	0.57	0.51	0.52	0.80
0.333	0.95	0.74	0.60	0.51	0.46	0.48	0.63
0.500	1.05	0.75	0.60	0.53	0.47	0.47	0.53
0.667	1.23	0.87	0.68	0.59	0.49	0.49	0.56
0.833	1.40	1.01	0.78	0.66	0.57	0.54	0.54
1.000	1.50	1.16	0.89	0.76	0.61	0.57	0.56

$f_{HP} = HP/HP_{max@rpm}$, $f_{RPM} = RPM/RPM_{max}$

Table 9-10. Fuel Consumption Map for Rotary SC Otto Engine (Reference 9-11)

f_{RPM}	f_{HP}						
	0.05	0.15	0.30	0.50	0.70	0.85	1.00
0.20	1.250	1.000	0.700	0.550	0.500	0.500	0.500
0.30	1.550	0.920	0.590	0.490	0.470	0.480	0.500
0.40	1.300	0.800	0.550	0.460	0.440	0.440	0.500
0.60	1.300	0.800	0.550	0.460	0.440	0.440	0.500
0.80	1.300	0.880	0.500	0.480	0.450	0.450	0.500
1.00	1.210	0.920	0.670	0.540	0.470	0.460	0.500

$f_{\text{HP}} = \text{HP}/\text{HP}_{\text{max@rpm}}$, $f_{\text{RPM}} = \text{RPM}/\text{RPM}_{\text{max}}$

Table 9-11. Fuel Consumption Map for FT Brayton Engine (Reference 9-24)

f_{RPM}	f_{HP}						
	0.10	0.20	0.30	0.50	0.70	0.90	1.00
0.10	0.972	0.911	0.911	0.951	1.022	1.133	1.214
0.20	0.759	0.698	0.688	0.708	0.759	0.810	0.799
0.40	0.557	0.476	0.466	0.486	0.516	0.557	0.546
0.60	0.526	0.455	0.445	0.445	0.455	0.466	0.455
0.80	0.648	0.496	0.455	0.445	0.455	0.455	0.445
1.00	0.982	0.734	0.557	0.511	0.506	0.491	0.481

$f_{\text{HP}} = \text{HP}/\text{HP}_{\text{max@rpm}}$, $f_{\text{RPM}} = \text{RPM}/\text{RPM}_{\text{max}}$

Table 9-12. Fuel Consumption Map for SS Brayton Engine (Reference 9-24)

f_{RPM}	f_{HP}						
	0.05	0.15	0.30	0.50	0.70	0.85	1.00
0.40	1.200	0.900	0.700	0.550	0.460	0.420	0.410
0.50	1.200	0.900	0.620	0.500	0.450	0.420	0.410
0.60	1.000	0.670	0.550	0.490	0.450	0.420	0.410
0.70	1.200	0.800	0.580	0.480	0.430	0.420	0.410
0.80	1.200	0.800	0.550	0.460	0.420	0.415	0.410
0.90	1.200	0.900	0.580	0.460	0.420	0.420	0.390
1.00	1.200	0.820	0.630	0.480	0.460	0.450	0.410

$f_{\text{HP}} = \text{HP}/\text{HP}_{\text{max@rpm}}$, $f_{\text{RPM}} = \text{RPM}/\text{RPM}_{\text{max}}$

Table 9-13. Fuel Consumption Map for Stirling Engine
(Metal) (Reference 9-27)

f_{RPM}	f_{HP}						
	0.05	0.20	0.35	0.50	0.65	0.80	1.00
0.168	0.620	0.530	0.462	0.420	0.396	0.380	0.369
0.263	0.562	0.435	0.380	0.368	0.360	0.356	0.350
0.421	0.525	0.425	0.383	0.375	0.370	0.368	0.366
0.579	0.530	0.460	0.413	0.396	0.391	0.389	0.386
0.737	0.570	0.496	0.458	0.435	0.425	0.420	0.418
0.895	0.650	0.577	0.522	0.475	0.472	0.467	0.466
1.000	0.745	0.650	0.578	0.539	0.510	0.509	0.506

$$f_{HP} = HP/HP_{\max@rpm}, f_{RPM} = RPM/RPM_{\max}$$

Table 9-14. Fuel Consumption Map for Stirling Engine (Ceramic)*

f_{RPM}	f_{HP}						
	0.05	0.20	0.35	0.50	0.65	0.80	1.00
0.168	0.428	0.387	0.369	0.358	0.350	0.345	0.339
0.263	0.372	0.354	0.342	0.335	0.330	0.327	0.328
0.421	0.362	0.340	0.328	0.322	0.318	0.316	0.314
0.579	0.365	0.345	0.336	0.331	0.326	0.322	0.316
0.737	0.380	0.356	0.346	0.342	0.338	0.335	0.333
0.895	0.528	0.402	0.368	0.355	0.349	0.346	0.344
1.000	0.760	0.495	0.405	0.372	0.360	0.358	0.355

$$f_{HP} = HP/HP_{\max@rpm}, f_{RPM} = RPM/RPM_{\max}$$

* Source: Fact-finding visit by ATSP team members to United Stirling of Sweden, Malmo, Sweden, Sept. 13, 1977.

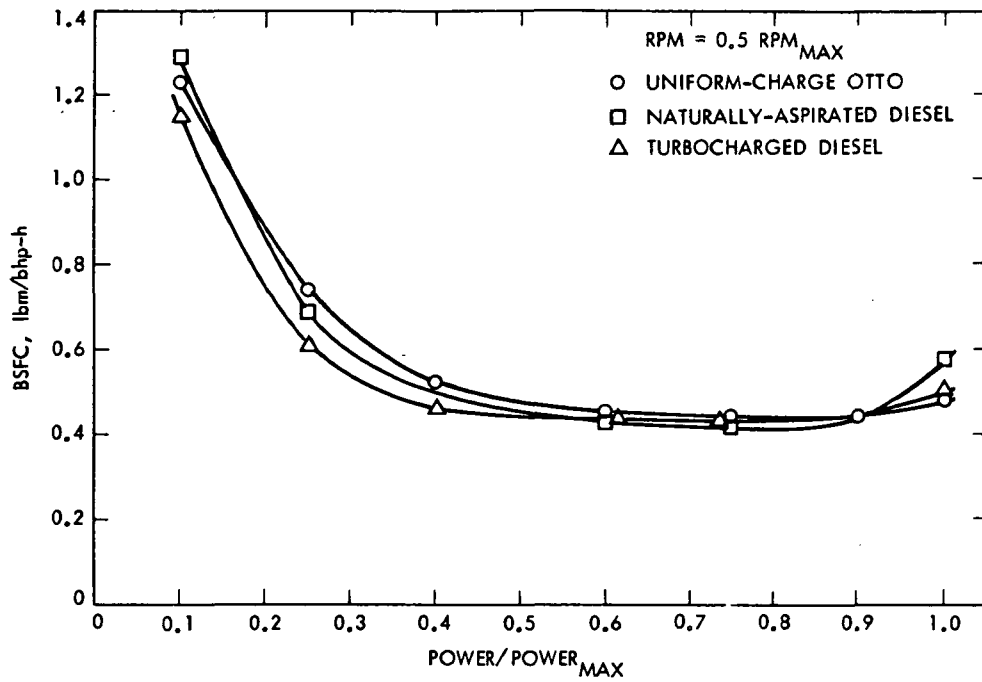


Figure 9-24. Fuel Consumption Characteristics for Conventional Engines

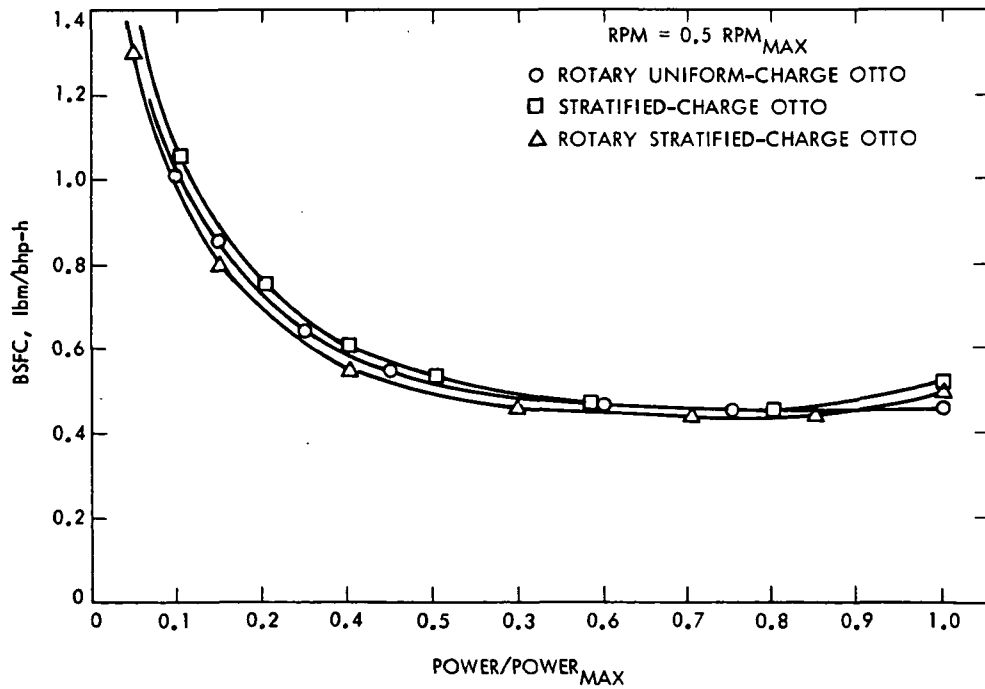


Figure 9-25. Fuel Consumption Characteristics for Advanced Conventional Engines

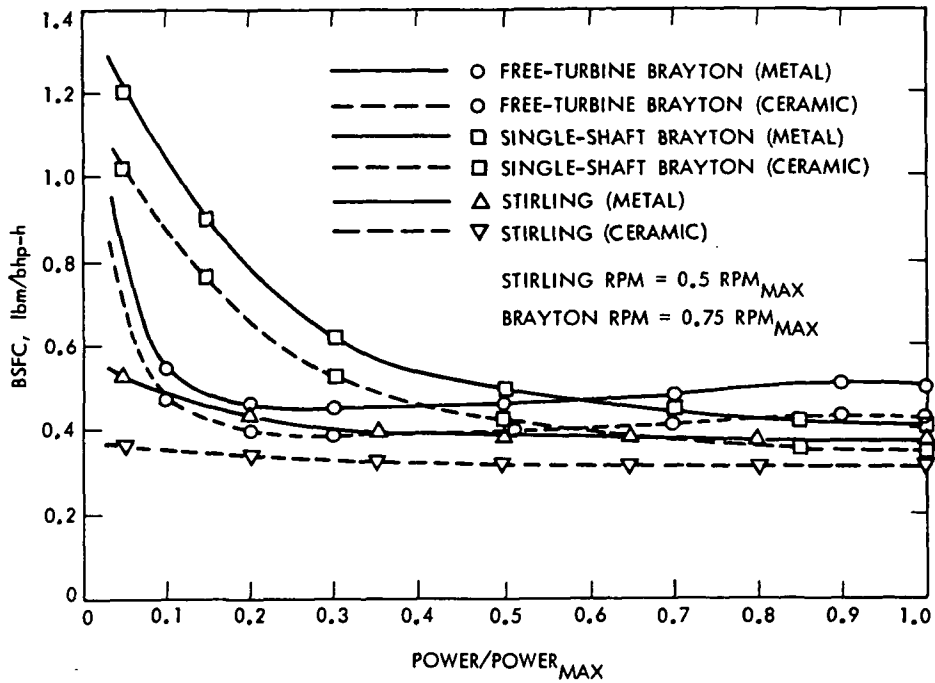


Figure 9-26. Fuel Consumption Characteristics for Advanced Alternative Engines

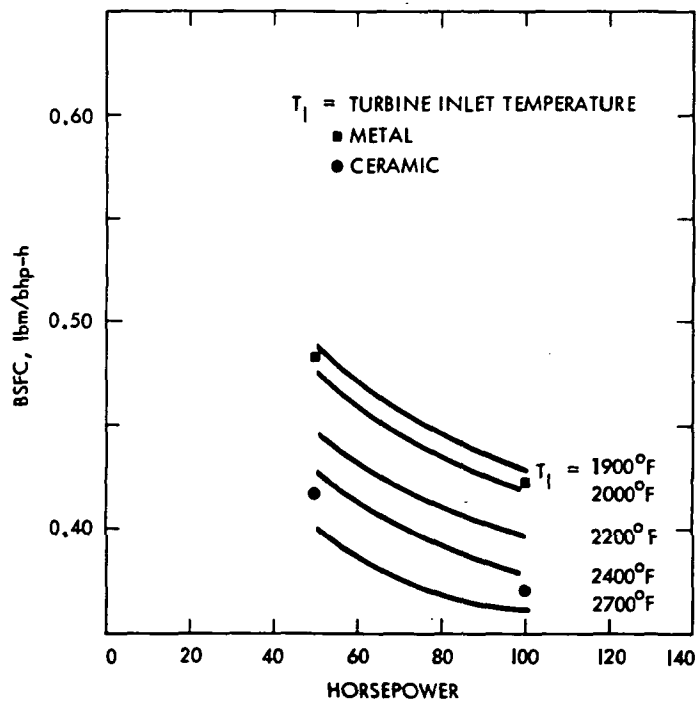


Figure 9-27. Effect of Turbine Inlet Temperature and Engine Size on Fuel Consumption of Brayton Engines

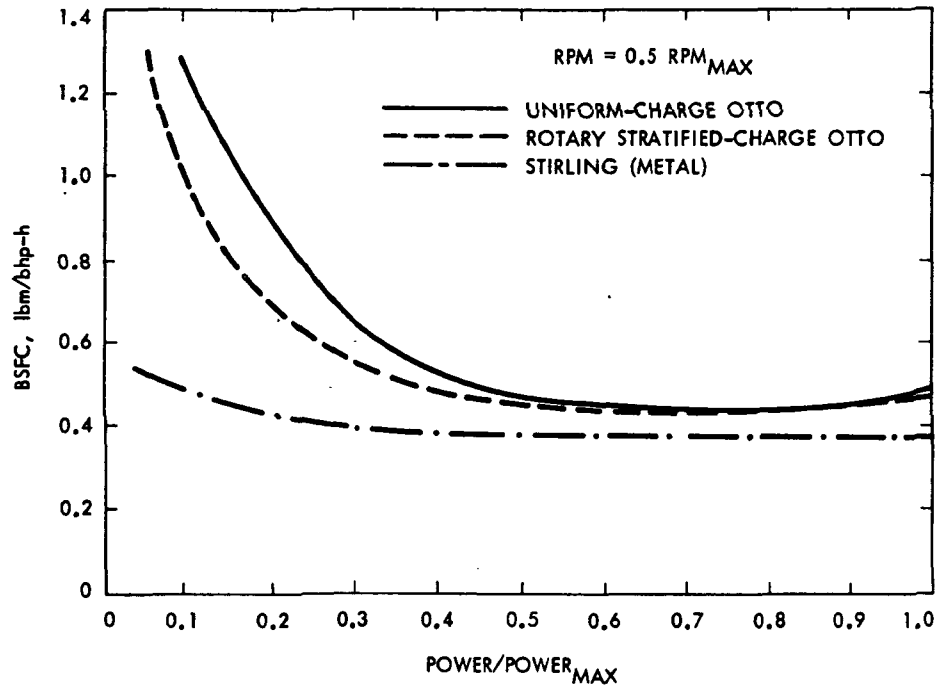


Figure 9-28. Comparison of Fuel Consumption Characteristics of Uniform-Charge Otto, Stratified-Charge Otto, and Stirling Engines

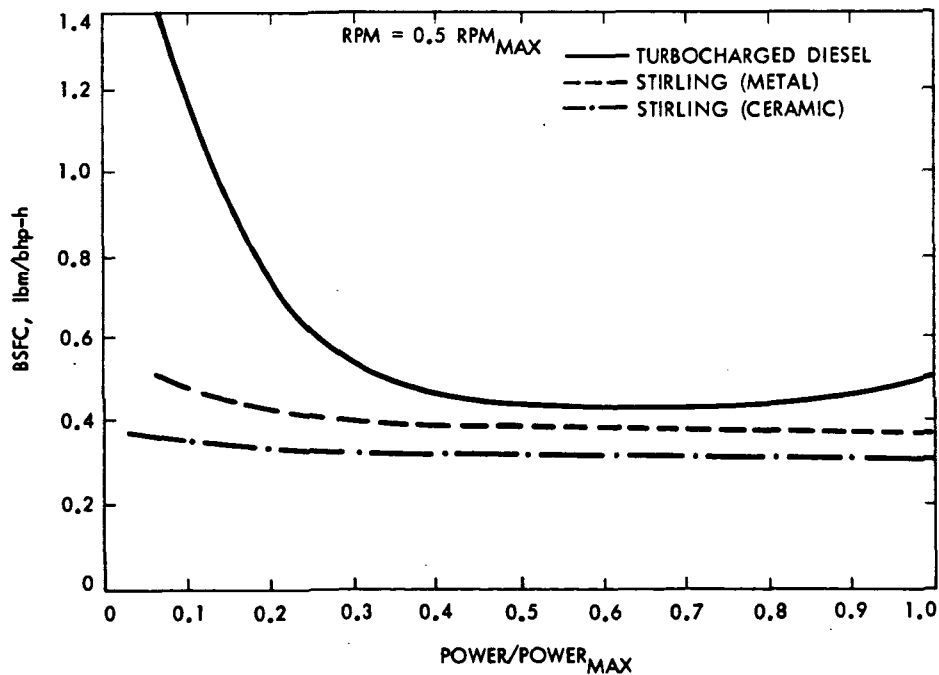


Figure 9-29. Comparison of Fuel Consumption Characteristics of Stirling and Diesel Engines

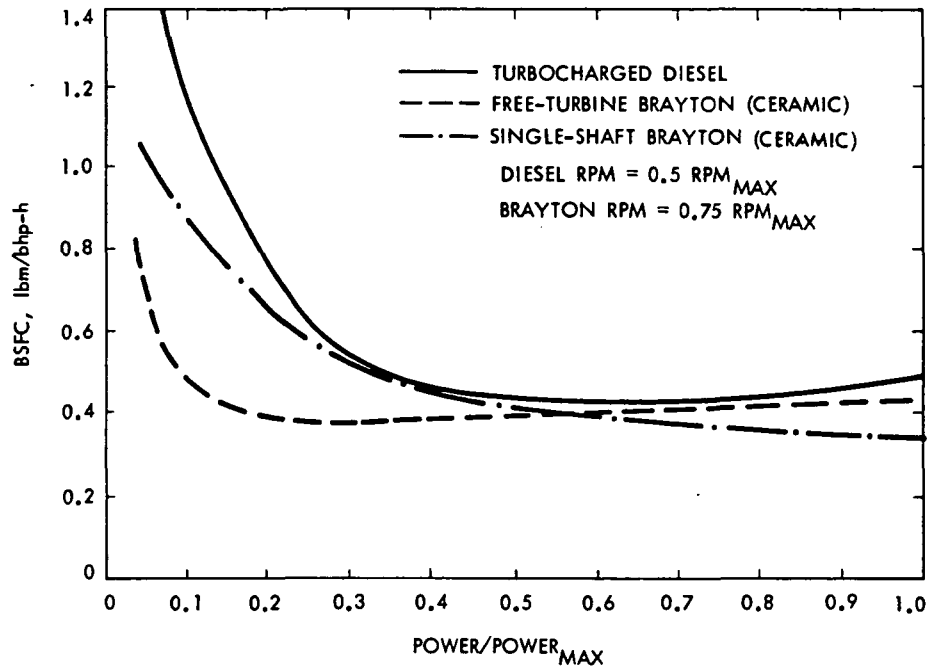


Figure 9-30. Comparison of Fuel Consumption Characteristics of Brayton and Diesel Engines

engine can be achieved in practice. The bsfc values of the metal and ceramic Stirling engines are compared with those of the TC diesel in Figure 9-29. Again the potential advantage of the Stirling engine is apparent. The excellent part-load bsfc of the Stirling engine is particularly attractive. The bsfc values of the ceramic FT Brayton and SS Brayton engines are compared with those of the TC diesel in Figure 9-30. Brayton engines show a potential for lower fuel consumption than the TC diesel; however, the margin is not as large as that for the ceramic Stirling, principally because of the better part-load bsfc of the Stirling.

Idle and deceleration fuel flow rates have a significant effect on the fuel economy which can be achieved with the various engines. This is especially true in city driving, as simulated by the EPA urban driving cycle, in which about 40 percent of the time is spent in either an idling or deceleration mode. The idle fuel flow rates which have been used in making fuel economy projections for the various engines are given in Figure 9-31. The low idle fuel flow of the diesel and the high idle fuel flow for the Brayton are particularly evident. The importance of reducing the idle fuel flow in the Brayton is well known, and considerable progress has been made in reducing it. However, it still remains higher than that for other engines.

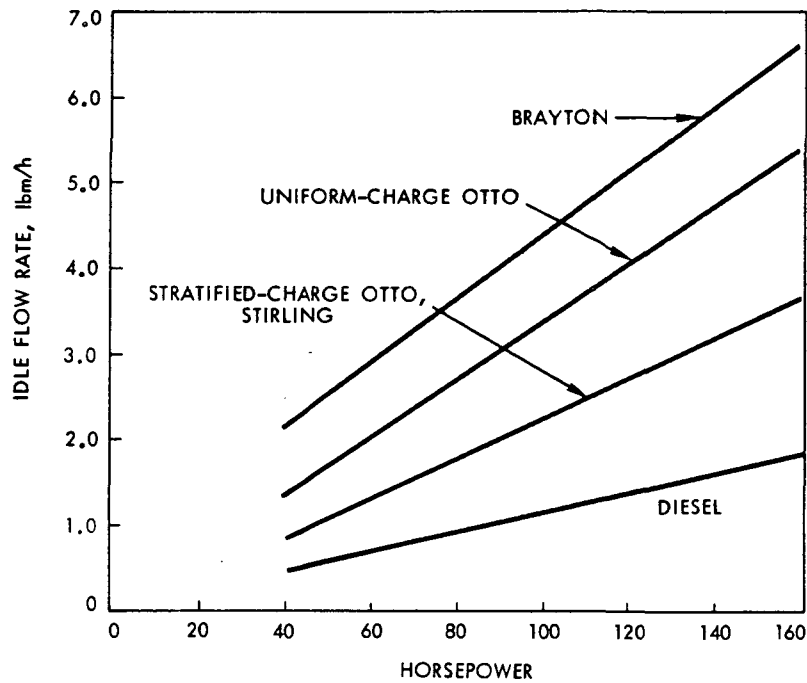


Figure 9-31. Idle Fuel Flow Rate Characteristics

Based on calculations made using a computer simulation program, it is possible to estimate the effect of idle fuel flow on vehicle fuel economy. The effect of the idle fuel flow on fuel economy over the EPA urban driving cycle can be written as

$$\frac{Y}{Y_0} = \frac{1}{1 - 0.0203 \frac{\omega_{i0} Y_0}{\rho_f} (1 - \omega_i / \omega_{i0})}$$

where

Y = fuel economy using idle fuel flow ω_i

Y_0 = fuel economy for the reference idle fuel flow ω_{i0} (lb/h)

ρ_f = fuel density (lb/gal)

The normalized fuel economy, Y/Y_0 , is shown in Figure 9-32 as a function of $\omega_{i0} Y_0 / \rho_f$ and ω_i / ω_{i0} . Consider the following example: $\omega_{i0} = 5$ lb/h, $Y_0 = 20$ mpg, $\rho_f = 7.3$ lb/gal, $\omega_i / \omega_{i0} = 0.5$. In this case, a 50 percent reduction in idle fuel flow results in about a 16 percent improvement in fuel economy on the urban cycle. The effect of idle and deceleration fuel flow on highway fuel economy is, on the other hand, negligible.

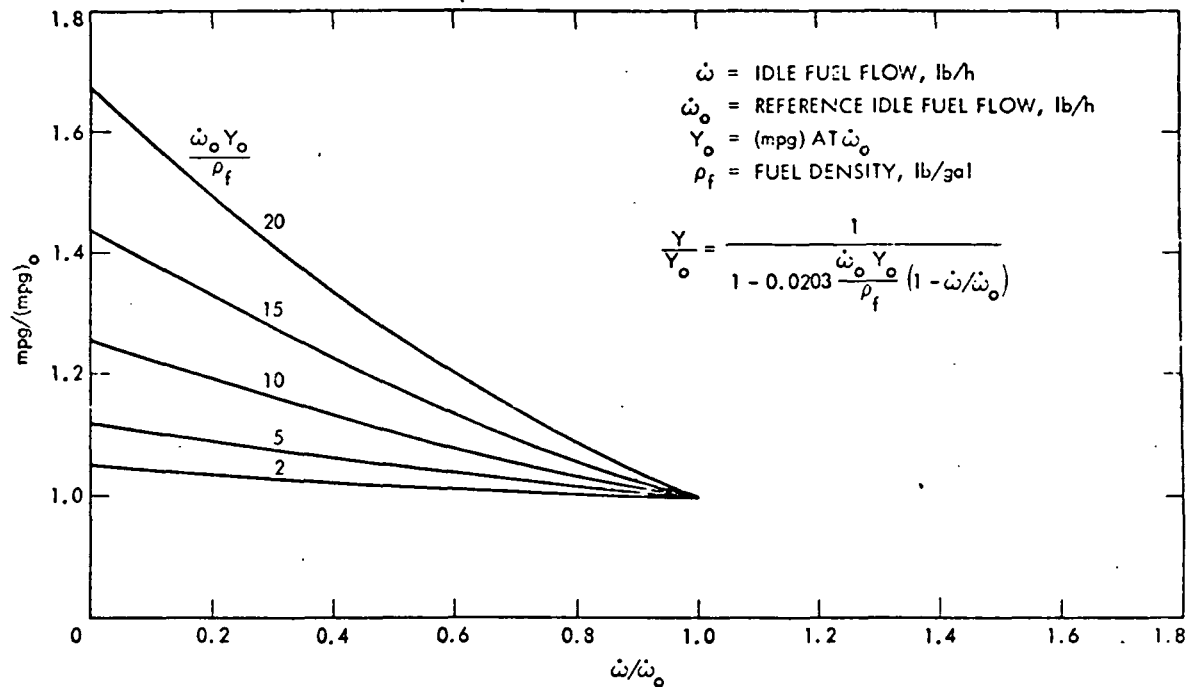


Figure 9-32. Effect of Idle Fuel Flow on Urban Cycle Fuel Economy

Other aspects of engine operation which are important for passenger car applications are engine transients (i.e., load variations) and accessory power requirements. Neither of these factors poses any significant problem for conventional engines. In fact, recent work on conventional engines has indicated that the effects of transients and accessories on vehicle fuel economy can be further reduced with the use of constant speed accessory drives or idle/deceleration fuel flow cut-off or modulation. DOE and the General Services Administration are jointly sponsoring an evaluation of a constant speed accessory drive to determine the potential fuel economy improvements which are possible with its use in passenger cars (Reference 9-30). The effect of load transients can be important for both Stirling and Brayton engines. For the Stirling engine, the following major transient effects are associated with the power level control system: (1) the power required to increase the working pressure for an increase in engine power and (2) the throttling loss which occurs when the working pressure is rapidly reduced for a decrease in engine power. For the gas turbine, the following major transient effects occur: (1) engine speed excursions into low efficiency portions of the component maps during vehicle accelerations and (2) high fuel flow and reductions in engine speed during decelerations. If steady-state engine dynamometer fuel consumption data is used, it is not possible to account for transient effects, and thus they are often not treated satisfactorily in making fuel economy projections.

9.6.2 Vehicle Fuel Economy

Following the methodology discussed in Section 9.2, Otto-Engine-Equivalent (OEE) vehicles were established for each power system. The engine weight and torque-speed characteristics representative of each engine type were used in these calculations. In establishing equivalent performance, the combined performance criteria were satisfied by minimizing the RMS deviation from baseline performances for the 0-60 mph acceleration time, 10-second acceleration distance, and 40-60 mph acceleration time.

The results of the OEE vehicle calculations are given in Table 9-15 for full-sized vehicles. The horsepower requirements for vehicles with the various engines are shown in Figure 9-33. Horsepowers are given as a delta from the baseline horsepower for the UC Otto engine. Vehicles with conventional engines (NA diesel, TC diesel) and advanced conventional engines (rotary UC Otto, SC Otto, rotary SC Otto) all require more horsepower than vehicles with the baseline (UC Otto) engine to achieve equivalent performance. Vehicles with the advanced alternative engines (FT Brayton, SS Brayton, Stirling) show definite advantages by requiring less horsepower than the baseline vehicle for equivalent performance. The corresponding vehicle weights for the OEE vehicles are given in Figure 9-34. Vehicles with FT Brayton and SS Brayton engines show definite weight advantages, being considerably lighter than the baseline vehicle. Vehicles with rotary engines also show some weight advantage, but not as much as the Brayton vehicles. Vehicles with NA diesel, TC diesel, and SC Otto engines are heavier than the baseline vehicle. The results of similar OEE vehicle calculations for small vehicles are given in Table 9-16. The engine horsepowers and vehicle weights for vehicles with the various engine types are shown in Figures 9-35 and 9-36, respectively. The trends shown in these figures are similar to those previously discussed for full-sized vehicles.

Using these OEE vehicle results and the representative fuel consumption maps which have been presented previously in this chapter for each engine type, vehicle fuel economies have been calculated using the vehicle computer simulation program. The computer results for the fuel economy of full-sized vehicles are given in Table 9-17. Fuel economy in mpg is shown for the city, highway, and composite driving cycles. Where the results have been calculated using diesel fuel, the mpg values are also shown in gasoline-equivalent mpg to allow a comparison of the results on an energy basis. Calculations are shown for both four-speed manual (M4) and five-speed manual (M5) transmissions with each engine type except the SS Brayton, which requires a continuously variable transmission (CVT). Results for vehicles with conventional engines are representative of 1978 technology. The emissions constraint considered for these calculations was the Federal legislated emission standards (0.4 g/mi HC, 3.4 g/mi CO, 1.0 g/mi NO_x).

Table 9-15. Otto-Engine-Equivalent Vehicle Results for Full-Sized Vehicles(1)

Power System	HP	Δ HP (2)	Curb Weight, lb	Δ (Weight), (2) lb
CONVENTIONAL ENGINES				
UC Otto	120	0	3200	0
NA diesel	140	20	3611	411
TC diesel	128	8	3349	149
ADVANCED CONVENTIONAL ENGINES				
Rotary UC Otto	123	3	3065	-135
SC Otto	138	18	3339	139
Rotary SC Otto	134	14	3139	-61
ADVANCED ALTERNATIVE ENGINES				
FT Brayton	91	-29	2741	-459
SS Brayton	88	-32	2675	-525
Stirling	110	-10	3192	-8
(1) Based on equivalent performance using the combined performance criteria.				
(2) Δ HP = HP - HP _{UC Otto} ; Δ (Weight) = Weight - Weight _{UC Otto} .				

These fuel economy projections are shown plotted in Figures 9-37, 9-38, and 9-39 for the city, highway, and composite driving cycles, respectively. It should be understood that the engine horsepowers and vehicle weights for the vehicles in this comparison differ since they have been calculated using the OEE vehicle concept. Fuel economies are all expressed as gasoline equivalent mileages. The conventional engines (UC Otto, NA diesel, TC diesel) and some of the advanced conventional engines (rotary UC Otto, SC Otto, rotary SC Otto) have one bar (cross-hatched) to represent 1978 values and another bar (open) to represent the projected 1985 values for these engines. The 1978 base for fuel economy is chosen to be the best fuel economy shown for vehicles with conventional engines.

On the composite driving cycle, vehicles with advanced alternative engines show significantly better fuel economy than vehicles with conventional engines when measured relative to the 1978 base. Relative to this base, advanced Brayton engines (ceramic) show 30 percent better fuel economy while advanced Stirling engines (ceramic) show a 40 percent

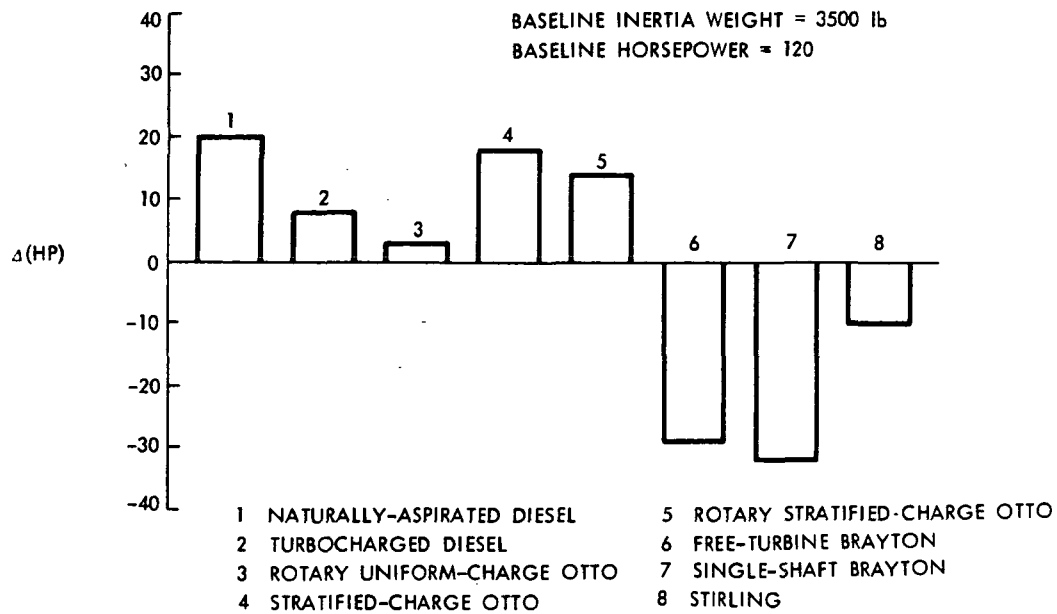


Figure 9-33. OEE Vehicle Horsepower for Full-Sized Vehicles

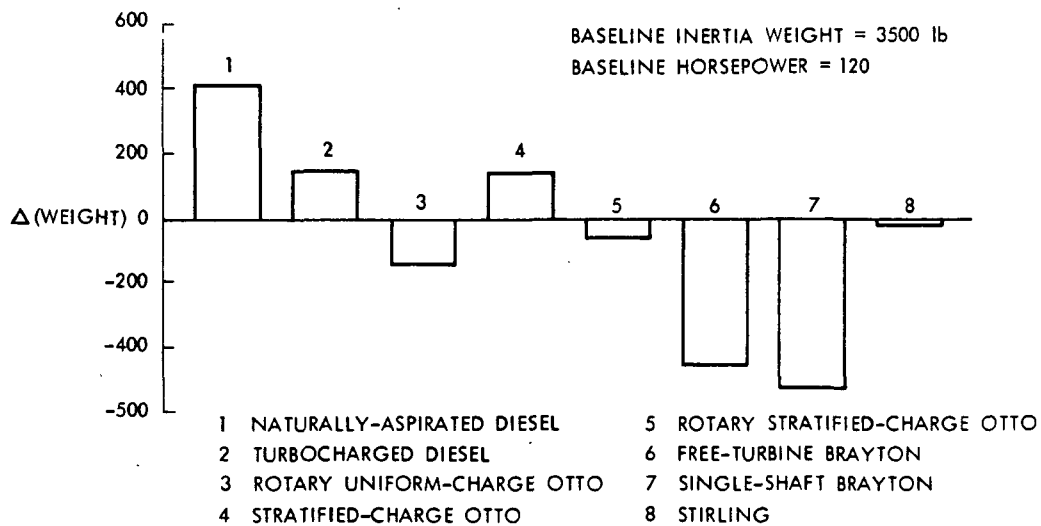


Figure 9-34. OEE Vehicle Weights for Full-Sized Vehicles

Table 9-16. Otto-Engine-Equivalent Vehicle Results for Small-Sized Vehicles⁽¹⁾

Power System	HP	HP ⁽²⁾	Curb Weight	(Weight) ⁽²⁾
CONVENTIONAL ENGINES				
UC Otto	60	0	1750	0
NA diesel	70	10	2003	253
TC diesel	67	7	1878	128
ADVANCED CONVENTIONAL ENGINES				
Rotary UC Otto	62	2	1681	-69
SC Otto	72	12	1888	138
Rotary SC Otto	67	7	1724	-26
ADVANCED ALTERNATIVE ENGINES				
FT Brayton	49	-11	1548	-202
SS Brayton	45	-15	1473	-277
Stirling	56	-4	1815	65

(1) Based on equivalent performance using the combine performance criteria.

(2) $HP = HP - HP_{UC\ Otto}$, $(Weight) = Weight - Weight_{UC\ Otto}$.

fuel economy advantage. Relative to the projected 1985 base, these fuel economy advantages of vehicles with Brayton and Stirling engines are reduced by about 10 percent. In looking at the plots of the city and highway results, it is clear that much of the fuel economy advantage of Brayton and Stirling vehicles comes from the highway cycle. The figures also show that vehicles with SC Otto and rotary engines are projected to have fuel economies between those for the UC Otto and TC diesel.

Since the time period for this study extends to the end of the century, it is important to assess the fuel economy impact of more stringent future emissions requirements, especially the 0.4 g/mi NO_x requirement. Projections of the comparative fuel economy for the various vehicles, using one possible set of future emissions standards (0.4 g/mi HC, 3.4 g/mi CO, 0.4 g/mi NO_x), are given in Figure 9-40 for a full-sized vehicle over the composite driving cycle. A fuel economy penalty has been imposed on vehicles with conventional and advanced conventional engines in order to meet the lower NO_x emissions requirement. The diesel engines have been dropped in this comparison since it

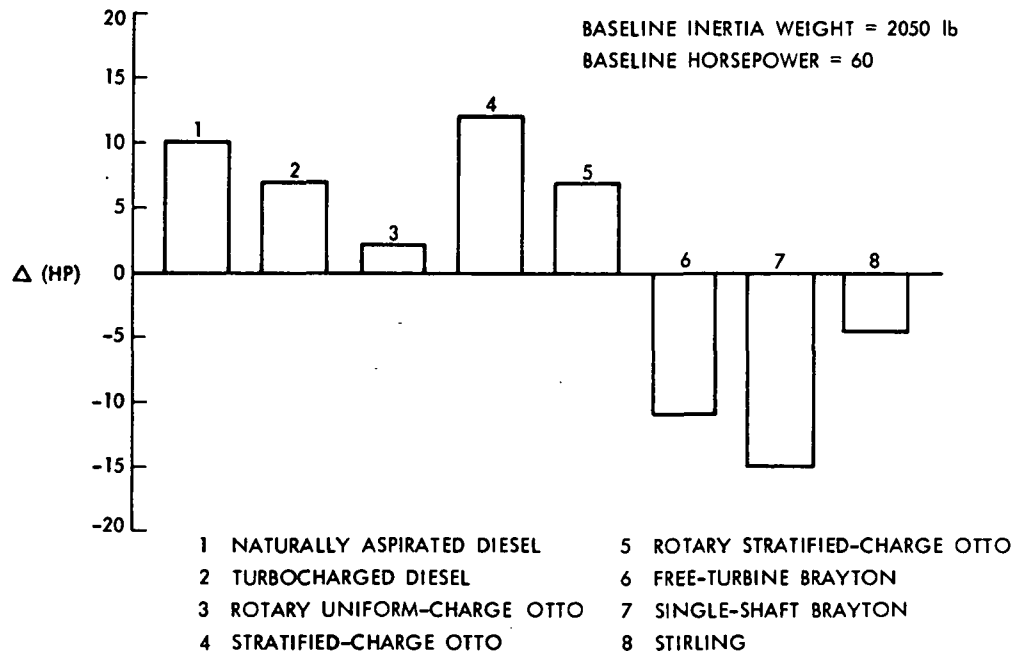


Figure 9-35. OEE Vehicle Horsepower for Small Vehicles

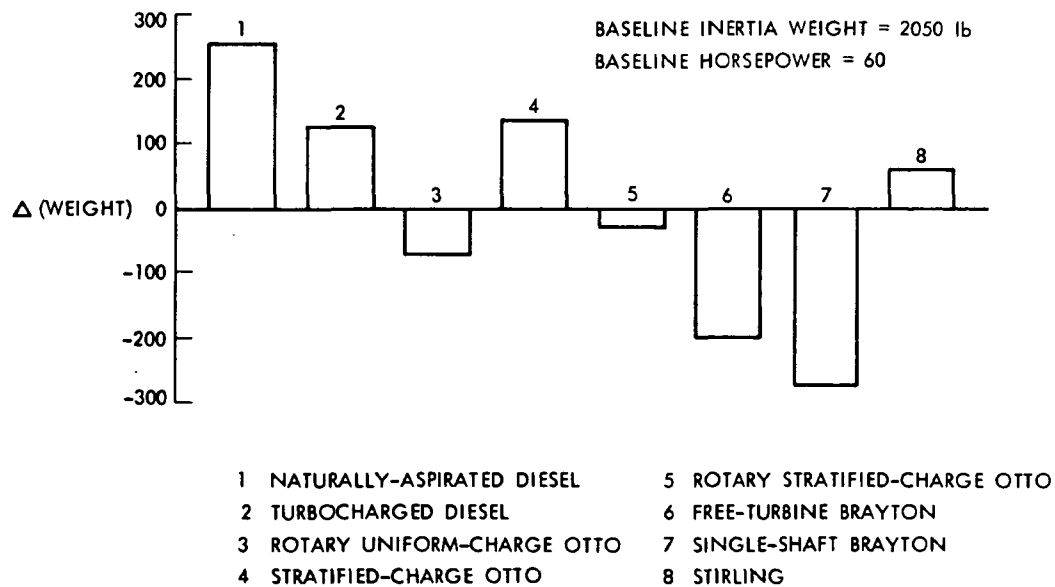


Figure 9-36. OEE Vehicle Weights for Small Vehicles

Table 9-17. Fuel Economy Projections for Full-Sized Vehicles
Using Various Engines (OEE Vehicles)

Power System	Transmission (1)	MPG - City		MPG - Highway		MPG - Composite	
		Diesel	Gasoline Equiv- alent	Diesel	Gasoline Equiv- alent	Diesel	Gasoline Equiv- alent
CONVENTIONAL ENGINES							
UC Otto	M4		15.1		23.8		18.1
	M5		16.6		30.0		20.8
NA Diesel	M4	22.8	20.1	25.5	22.5	23.9	21.1
	M5	24.9	22.0	31.7	28.0	27.6	24.3
TC Diesel	M4	20.7	18.3	31.4	27.7	24.5	21.6
	M5	21.9	19.4	35.8	31.5	26.6	23.4
ADVANCED CONVENTIONAL ENGINES							
Rotary UC Otto	M4		16.9		26.6		20.2
	M5		17.6		29.6		21.5
SC Otto	M4		14.7		23.6		17.7
	M5		15.5		27.7		19.3
Rotary SC Otto	M4		17.0		27.8		20.6
	M5		17.7		31.5		22.1

Table 9-17. Fuel Economy Projections for Full-Sized Vehicles
Using Various Engines (OEE Vehicles) (Continuation 1)

Power System	Transmission ⁽¹⁾	MPG - City		MPG - Highway		MPG - Composite	
		Diesel	Gasoline Equiv- alent	Diesel	Gasoline Equiv- alent	Diesel	Gasoline Equiv- alent
ADVANCED ALTERNATIVE ENGINES							
FT Brayton (metal)	M4	23.0	20.3	45.9	40.5	29.7	26.2
	M5	21.0	18.5	44.1	39.0	27.5	24.3
FT Brayton (ceramic)	M4	25.5	22.5	52.0	45.9	33.1	29.2
	M5	23.3	20.6	50.0	44.2	30.7	27.1
SS Brayton (metal)	CVT	22.5	19.8	42.1	37.2	28.5	25.1
SS Brayton (ceramic)	CVT	24.9	22.0	47.8	42.2	31.8	28.0
Stirling (metal)	M4		22.6		38.7		27.8
	M5		22.8		42.7		28.8
Stirling (ceramic)	M4		25.0		48.1		31.9
	M5		24.2		50.1		32.3

(1) M4 - four-speed manual, M5 - five-speed manual, CVT - continuously variable transmission

⁽¹⁾ M4 - four-speed manual, M5 - five-speed manual, CVT - continuously variable transmission

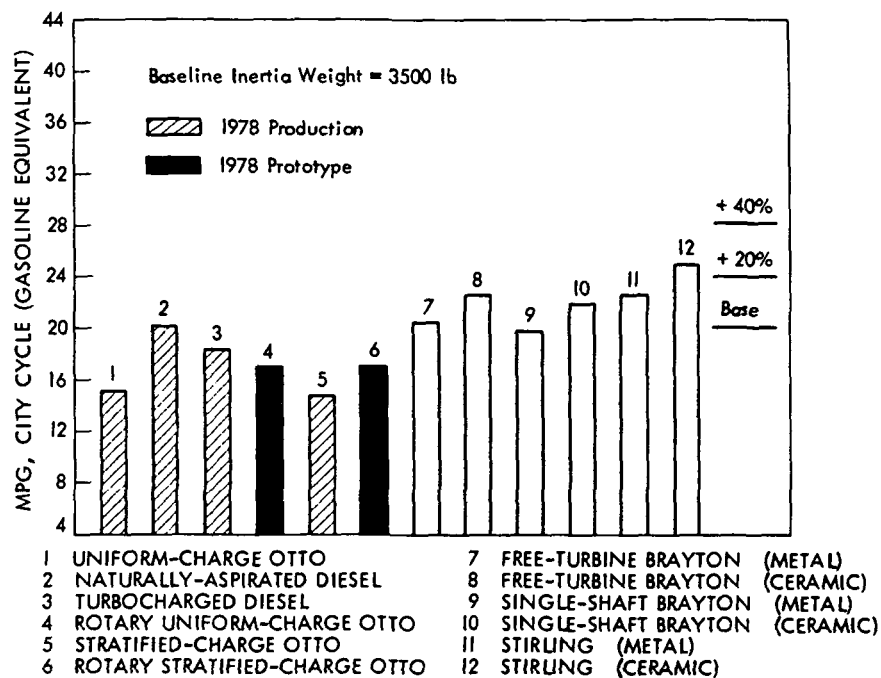


Figure 9-37. City Fuel Economy Comparison for Full-Sized OEE Vehicles (1.0 g/mi NO_x)

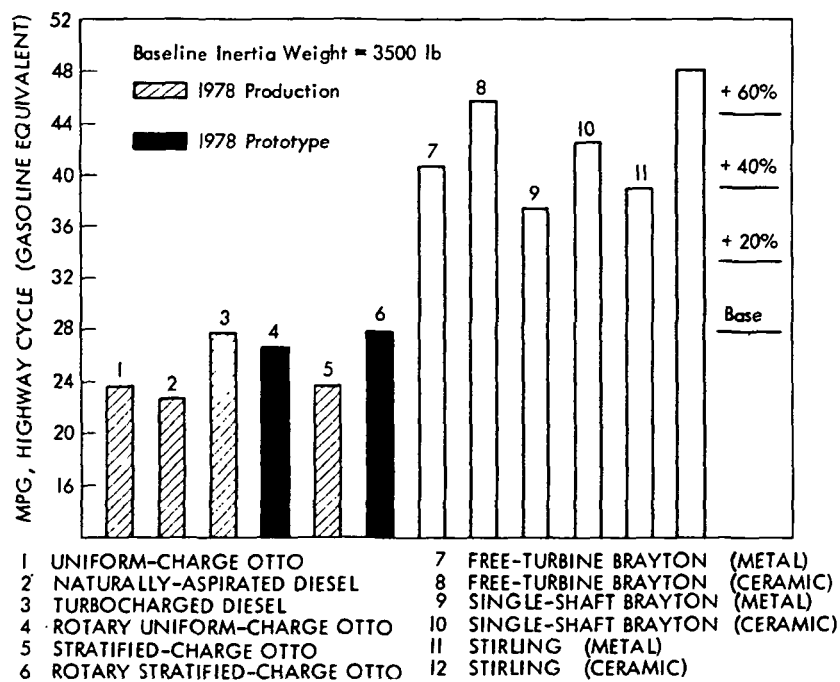


Figure 9-38. Highway Fuel Economy Comparison for Full-Sized OEE Vehicles (1.0 g/mi NO_x)

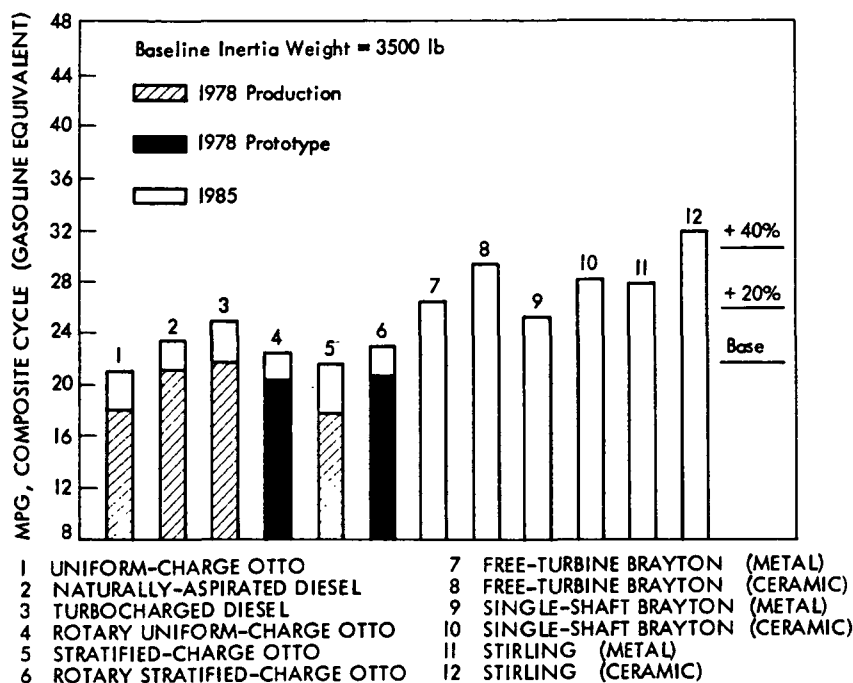


Figure 9-39. Composite Fuel Economy Comparison for Full-Sized OEE Vehicles (1.0 g/mi NO_x)

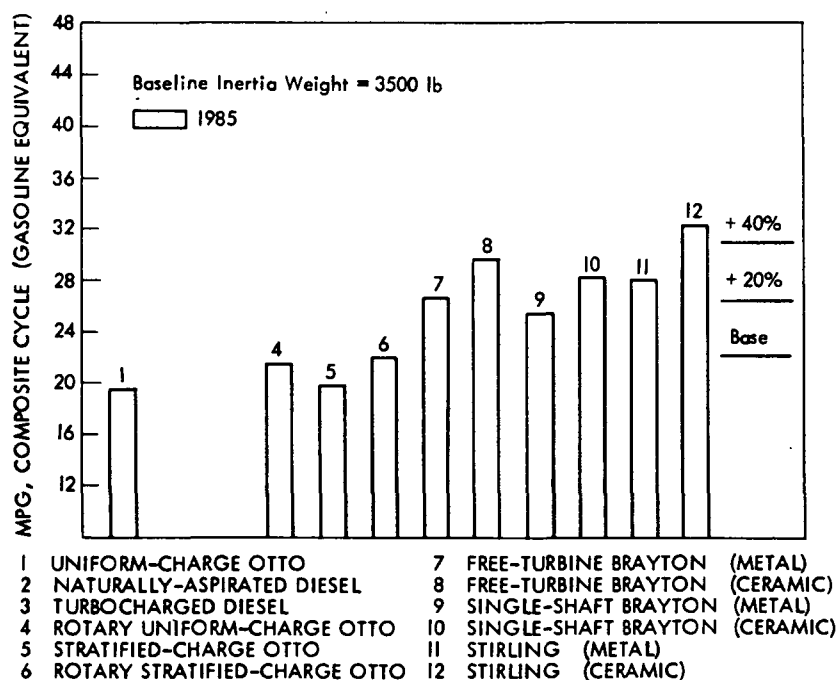


Figure 9-40. Composite Fuel Economy Comparison for Full-Sized OEE Vehicles (0.4 g/mi NO_x)

is not expected that the diesel can meet this NO_x requirement in full-sized vehicles. Vehicles with advanced alternative engines (Brayton and Stirling) are expected to meet these emissions levels with no penalty in fuel economy. Relative to the 1985 base, the advanced Brayton engines (ceramic) show 30 percent better fuel economy, and the advanced Stirling engines (ceramic) show 40 percent better fuel economy advantage.

Again using the OEE vehicle concept, computer results for the fuel economy of small vehicles are given in Table 9-18. Plots of these fuel economy projections are shown in Figures 9-41, 9-42, and 9-43 for the city, highway, and composite driving cycle, respectively. Both 1978 and 1985 bases are shown for vehicles with conventional engines. On the composite driving cycle, vehicles with Stirling engines show significantly better fuel economy than vehicles with conventional engines. Relative to the 1978 base, advanced Stirling engines (ceramic) show a 40 percent fuel economy advantage. In this small-sized vehicle, vehicles with Brayton engines do not show the same fuel economy advantage which they show for full-sized vehicles due to their size-related penalties and relatively poor idle and part-load fuel consumption. Again vehicles with SC Otto and rotary engines are projected to have fuel economies between those for the UC Otto and TC diesel. It is expected that the UC Otto engine with the three-way catalyst emission control system can be calibrated to meet the 0.4 g/mi NO_x emissions requirement in this weight vehicle with no fuel economy penalty. Thus, essentially the same comparison would exist between the advanced alternative engines and the conventional engines for the more strict emissions requirements.

Sensitivity studies have been made to determine the effect of different performance criteria on the fuel economy projections. The fuel economy projections which have been presented were based on equivalent vehicle performance using the combined performance criteria. This criteria was satisfied by minimizing the RMS deviation from baseline performance for the 0-60 mph acceleration time, 10-second acceleration distance, and 40-60 mph acceleration time. OEE vehicles were established by satisfying each performance criteria separately and calculating the vehicle fuel economy. The results of these calculations are given in Table 9-19 for a full-sized vehicle powered by a TC diesel engine. The fuel economy projections based on the four performance criteria are all contained within a ± 2 percent band. This indicates that the performance criterion selected has a small effect on the fuel economy projections, at least for the performance criterion evaluated.

It is important to examine the sensitivity of these fuel economy projections to other assumptions made in the OEE vehicle concept which was used in this study. Two of these key items are engine weights and engine transient response. The engine weight characteristics used to represent Brayton and Stirling engines are projections of what should be achieved with these engines as they become more developed; however, most current developmental engines fall short of this goal. Engine weight becomes even more important since the calculation procedure for an OEE vehicle includes vehicle weight

Table 9-18. Fuel Economy Projections for Small Vehicles
Using Various Engines (OEE Vehicles)

Power System	Transmission (1)	MPG - City		MPG - Highway		MPG - Composite	
		Diesel	Gasoline Equiv- alent	Diesel	Gasoline Equiv- alent	Diesel	Gasoline Equiv- alent
		CONVENTIONAL ENGINES					
UC Otto	M4		29.5		41.1		33.8
	M5		31.4		46.3		36.7
NA Diesel	M4	40.3	35.6	46.1	40.6	42.7	37.7
	M5	43.2	38.1	53.4	47.1	47.2	41.7
TC Diesel	M4	37.7	33.3	53.0	46.7	43.3	38.2
	M5	39.2	34.6	56.8	50.1	45.5	40.2
ADVANCED CONVENTIONAL ENGINES							
Rotary UC Otto	M4		31.7		44.1		36.3
	M5		32.6		47.2		37.9
SC Otto	M4		27.5		39.8		31.9
	M5		28.7		44.8		34.2
Rotary SC Otto	M4		32.2		46.2		37.3
	M5		33.0		49.6		38.9

Table 9-18. Fuel Economy Projections for Small Vehicles Using Various Engines (OEE Vehicles) (Continuation 1)

Power System	Transmission	(1)	MPG - City		MPG - Highway		MPG - Composite	
			Diesel	Gasoline Equivalent	Diesel	Gasoline Equivalent	Diesel	Gasoline Equivalent
			ADVANCED ALTERNATIVE ENGINES					
FT Brayton (metal)	M4		36.8	32.5	56.1	49.5	43.6	38.4
	M5							
FT Brayton (ceramic)	M4		43.9	38.8	64.5	56.9	51.3	45.2
	M5							
SS Brayton	CVT		35.3	31.1	52.6	46.4	41.4	36.6
(metal)								
SS Brayton	CVT		39.5	34.9	59.8	52.8	46.7	41.2
(ceramic)								
Stirling	M4			41.7		53.5		46.3
(metal)	M5			42.2		59.9		48.7
Stirling	M4			47.2		67.4		54.6
(ceramic)	M5			47.1		71.3		55.6

(1) M4 - four-speed manual, M5 - five-speed manual, CVT - continuously variable transmission

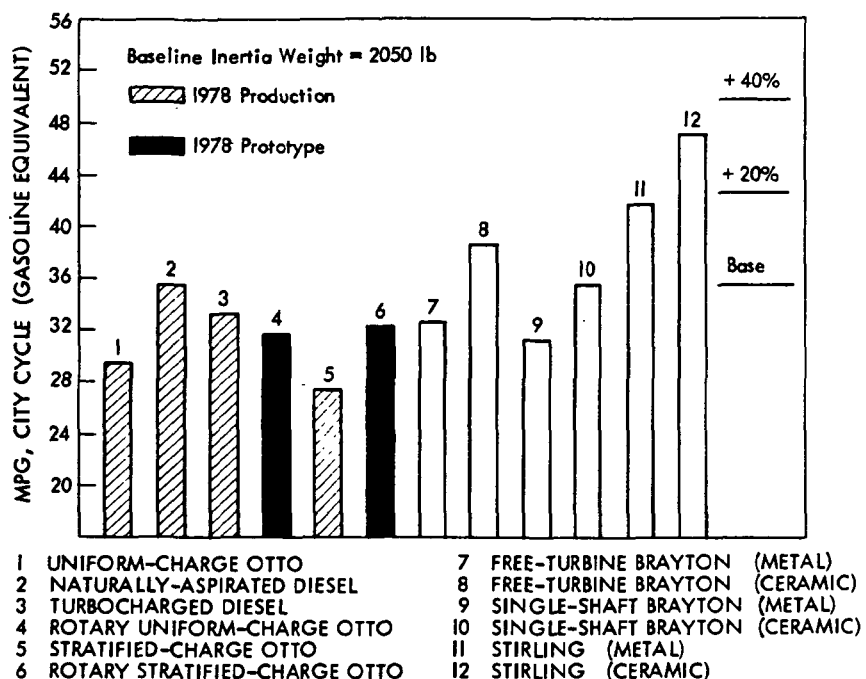


Figure 9-41. City Fuel Economy Comparison for Small OEE Vehicles (1.0 g/mi NO_x)

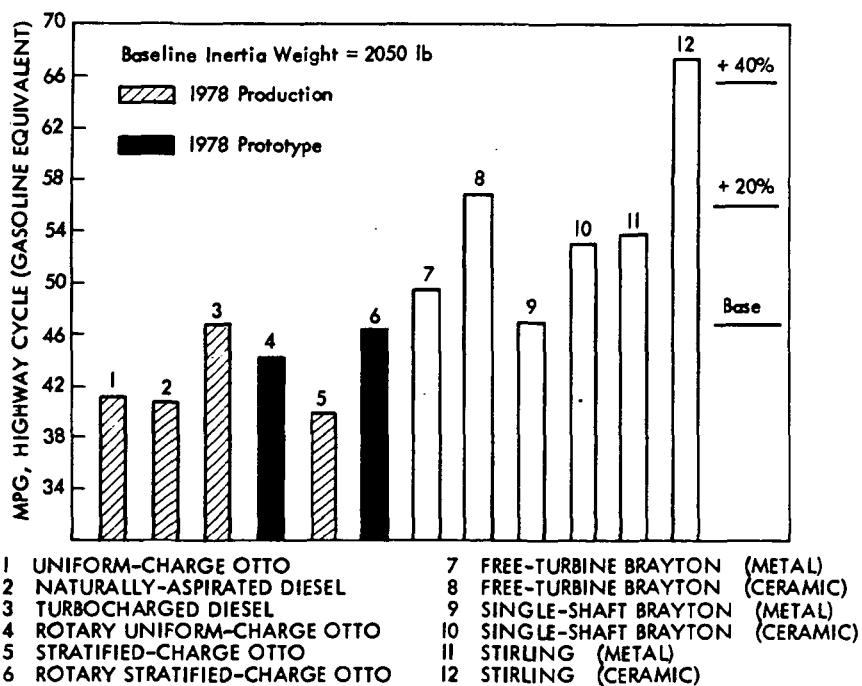


Figure 9-42. Highway Fuel Economy Comparison for Small OEE Vehicles (1.0 g/mi NO_x)

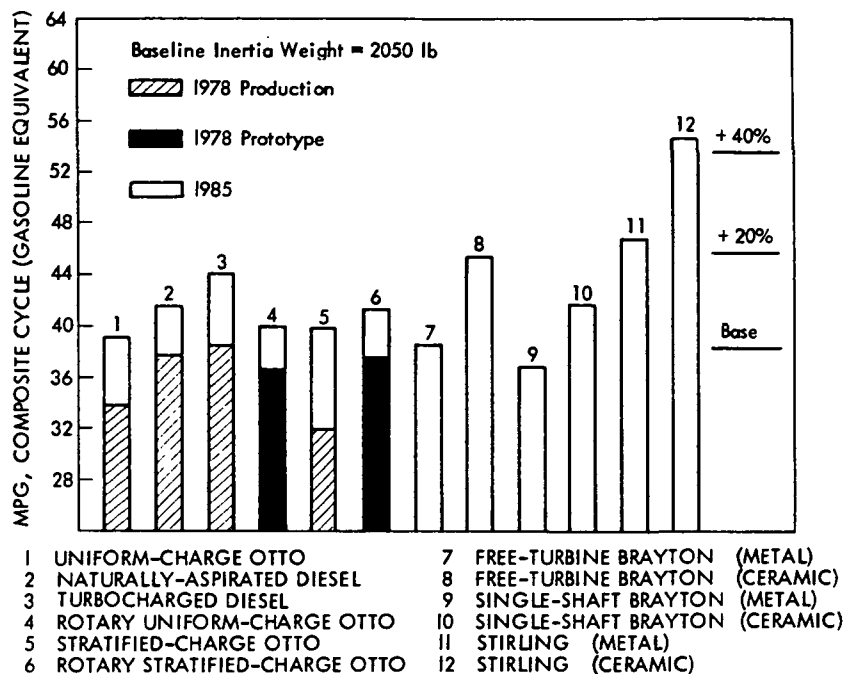


Figure 9-43. Composite Fuel Economy Comparison for Small OEE Vehicles (1.0 g/mi NO_x)

Table 9-19. Effect of Performance Criteria on TC Diesel Fuel Economy in Full-Sized Vehicles

Performance Criteria	MPG - Composite Cycle	
	Diesel	Gasoline Equivalent
Combined Criteria	24.5	21.6
0 - 60 mph Time	24.3	21.4
10 - Second Distance	24.1	21.2
40 - 60 mph Time	25.0	22.0

propagation effects to establish alternative vehicles which are designed with the same degree of optimization. Also, in sizing the engines to provide equivalent performance, engine transient response has been neglected. If acceleration time is measured relative to the time when the accelerator pedal is depressed, any delayed response of the engine would be included in the acceleration time. This factor could have an effect on sizing the engines for equivalent vehicle performance. To investigate these effects, fuel economy calculations have been made with all engine types having the same engine horsepower and same vehicle weight. These fuel economy results are given in Tables 9-20 and 9-21 for full-sized and small vehicles, respectively.

The fuel economy projections for full-sized vehicles are given in Figure 9-44 for the composite cycle. On this basis, the advanced alternative engines (Brayton and Stirling) show less fuel economy advantage over vehicles with conventional engines. This emphasizes the importance of developing the alternative engines to meet the engine weight goals and have transient response characteristics at least comparable to conventional engines.

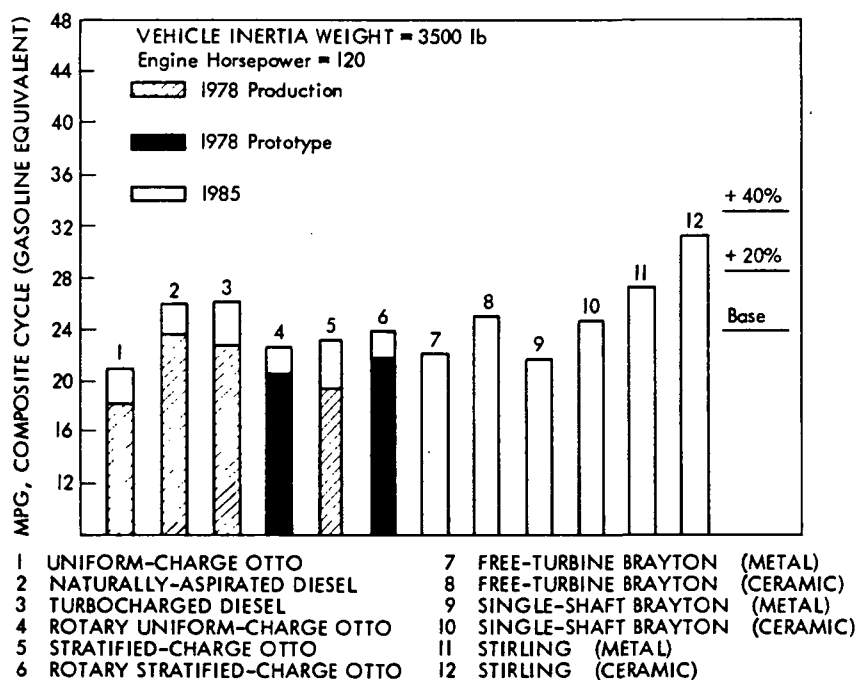


Figure 9-44. Composite Fuel Economy Comparison for Full-Sized Non-OEE Vehicles (1.0 g/mi NO_x)

Table 9-20. Fuel Economy Projections for Full-Sized Vehicles
Using Various Engines (Not OEE Vehicles)(1)

Power System	Transmission (2)	MPG - City		MPG - Highway		MPG - Composite	
		Diesel	Gasoline Equivalent	Diesel	Gasoline Equivalent	Engine	Gasoline Equivalent
		CONVENTIONAL ENGINES					
UC Otto	M4		15.1		23.8		18.1
NA Diesel	M4	25.1	22.1	28.9	25.5	26.7	23.5
TC Diesel	M4	21.5	18.9	32.9	29.0	25.4	22.5
ADVANCED CONVENTIONAL ENGINES							
Rotary UC Otto	M4		16.9		26.7		20.2
SC Otto	M4		15.9		25.7		19.2
Rotary SC Otto	M4		17.7		29.2		21.5

Table 9-20. Fuel Economy Projectons for Full-Sized Vehicles Using Various Engines (Not OEE Vehicles)(1) (Continuation 1)

Power System	Transmission (2)	MPG - City		MPG - Highway		MPG - Composite	
		Diesel	Gasoline Equiv- alent	Diesel	Gasoline Equiv- alent	Engine	Gasoline Equiv- alent
		ADVANCED ALTERNATIVE ENGINES					
FT Brayton (metal)	M4	18.9	16.6	40.9	36.1	24.9	22.0
FT Brayton (ceramic)	M4	21.1	18.6	46.7	41.2	28.0	24.7
SS Brayton (metal)	CVT	19.1	16.9	37.1	32.8	24.5	21.6
SS Brayton (ceramic)	CVT	21.5	18.9	42.4	37.4	27.6	24.3
Stirling (metal)	M4		21.9		37.8		27.0
Stirling (ceramic)	M4		24.2		47.1		31.0

(1) All vehicles have inertia weights of 3500 lb and engine horsepower of 120

(2) M4 - four-speed manual, CVT - continuously variable transmission

(1) All vehicles have inertia weights of 3500 lb and engine horsepower of 120

(2) M4 - four-speed manual, CVT - continuously variable transmission

Table 9-21. Fuel Economy Projections for Small Vehicles
Using Various Engines (Not OEE Vehicles)(1)

Power Systems	Transmission (2)	MPG - City		MPG - Highway		MPG - Composite	
		Diesel	Gasoline Equiv- alent	Diesel	Gasoline Equiv- alent	Diesel	Gasoline Equiv- alent
CONVENTIONAL ENGINES							
UC Otto	M4		29.5		41.1		33.8
NA Diesel	M4	44.4	39.2	51.1	45.1		41.6
TC Diesel	M4	39.8	35.1	56.5	49.8		40.5
ADVANCED CONVENTIONAL ENGINES							
Rotary UC Otto	M4		31.7		44.3		36.4
SC Otto	M4		30.4		43.6		35.2
Rotary SC Otto	M4		33.1		48.0		38.5

Table 9-21. Fuel Economy Projections for Small Vehicles Using Various Engines (Not OEE Vehicles)(1) (Continuation 1)

Power Systems	Transmission (2)	MPG - City		MPG - Highway		MPG - Composite	
		Diesel	Gasoline Equivalent	Diesel	Gasoline Equivalent	Diesel	Gasoline Equivalent
ADVANCED ALTERNATIVE ENGINES							
FT Brayton (metal)							
FT							
SS Brayton (metal)	CVT	30.9	27.3	49.0	43.3		32.7
SS Brayton (ceramic)	CVT	34.9	30.8	56.0	49.4		37.1
Stirling (metal)							
Stirling (ceramic)							

(1) All vehicles have inertia weights of 2050 lb and engine horsepower of 60

(2) M4 - four-speed manual, CVT - continuously variable transmission

(1) All vehicles have inertia weights of 2050 lb and engine horsepower of 60

(2) M4 - four-speed manual, CVT - continuously variable transmission

The need to improve air quality has led to stringent Federal emissions standards for controlling exhaust emissions from automobiles. The oil embargo by OPEC nations has helped focus the need for energy conservation in this country, especially in the use of petroleum for transportation. This need to conserve petroleum has led to the passage of Federal fuel economy standards and has led to an easing of emissions standards for passenger cars. Up through 1985, the most stringent Federal emissions standards are set at (0.4 g/mi HC, 3.4 g/mi CO, 1.0 g/mi NO_x) with the former 0.4 g/mi NO_x standard being retained as a research goal. Because of its special pollution problems, California requires that vehicles meet the (0.4 g/mi HC, 9.0 g/mi CO, 0.4 g/mi NO_x) values starting in 1982. Consideration is being given to waiving the 0.4 g/mi NO_x requirement and instead requiring vehicles to be certified for 100,000 miles (currently a 50,000 mile certification is required).

In addition to changes in Federal legislation, the successful introduction of oxidation and three-way catalysts on the conventional UC Otto engines has significantly reduced the need for low untreated emissions from these engines. The combined effect of technical and legislative developments has focused attention on meeting the Federal legislated emissions standards with as high a vehicle fuel economy as possible. In this long-range study, each engine type is evaluated for its ability to meet not only the current legislated emissions standards but also possible future requirements (0.4 g/mi HC, 3.4 g/mi CO, 0.4 g/mi NO_x). Also, the need for controlling presently unregulated emissions, such as particulates, sulfates, and odor will continue to receive attention and must be considered in assessing the emissions potential of vehicles powered by the various engines.

A summary of vehicle emissions test data for vehicles with conventional engines (UC Otto, turbocharged UC Otto, NA diesel, TC diesel) and advanced conventional engines (SC Otto, rotary UC Otto) is given in Table 9-22. This table includes advanced prototype engines and production engines with advanced emissions control systems. Similar test data for vehicles with advanced alternative engines (Brayton and Stirling) is given in Table 9-23. Unfortunately, little vehicle emissions data is available for vehicles with Brayton or Stirling engines, especially ones having combustor/engine control systems sufficiently optimized to control emissions to the statutory standards. Thus, in many cases their emissions potential must be inferred from steady-state combustor data and a general knowledge of pollutant formation during combustion.

The emissions data for vehicles with the various engine types is plotted in Figures 9-45, 9-46, and 9-47. Boundary lines representing the Federal legislated emission standards (0.4 g/mi HC, 3.4 g/mi CO, 1.0 g/mi NO_x) and the possible future emissions requirement (0.4 g/mi HC, 3.4 g/mi CO, 0.4 g/mi NO_x) are given in the figures to aid in evaluating each engine type.

Table 9-22. Vehicle Emissions Test Data for Conventional Engines and Advanced Conventional Engines

Engine(1) Type	Vehicle Manufacturer	Inertia Weight	Engine Type/HP	Mixture Prep.(1)	Emission Control		Emissions, g/mi		
					Catalyst	EGR	HC	CO	NO _x
CONVENTIONAL ENGINES									
UC, Otto, NA	GM	3000	L4/85	Carb	3-way	Yes	0.38	5.7	1.20
	Volvo	3000	L4/102	FI	3-way	None	0.20	2.7	0.18
	Volvo	3500	V6/130	FI	3-way	None	0.35	3.6	0.85
	Saab	3000	L4/110	FI	3-way	None	0.21	3.9	0.14
UC, Otto, TC	Saab	3000	L4/130	FI	3-way	None	0.23	2.5	0.74
NA Diesel	VW	2250	L4/50	FI	None	None	0.30	1.0	1.05
	VW	2500	L4/50	FI	None	None	0.50	1.0	1.15
	Peugeot	3500	L4/70	FI	None	None	0.65	1.6	1.10
	Mercedes-Benz	3500	L4/62	FI	None	None	0.35	1.1	1.15
	Mercedes-Benz	4000	L5/80	FI	None	None	0.10	1.0	1.70
TC Diesel	VW	2250	L4/70	FI	None	None	0.22	0.98	0.93
	VW	3000	L4/70	FI	None	None	0.14	0.8	1.20
	VW	3000	L4/70	FI	None	Yes	0.17	1.6	0.50

Table 9-22. Vehicle Emissions Test Data for Conventional Engines and Advanced Conventional Engines (Continuation 1)

Engine(1) Type	Vehicle Manufacturer	Inertia Weight	Engine Type/HP	Mixture Prep.(1)	Emission Control Catalyst	Emissions, g/ml		
						HC	CO	NO _x
ADVANCED CONVENTIONAL ENGINES								
SC Otto, PC	Honda CVCC	2000	L4/60	Carb	None	0.30	2.7	1.2
	Honda CVCC	2500	L4/68	Carb	None	0.28	2.4	1.2
SC Otto, DI	Ford Proco(2)	5000	V8/	FI	Oxidation	0.22	0.10	0.78
Rotary UC	NSU(2)	3000	170	Carb	Oxidation	0.30	0.60	0.65
Otto	Toyota Kogyo(2)	2750	135	Carb	Oxidation	0.18	1.0	0.80

(1) NA - naturally aspirated, TC - turbocharged, PC - pre-chamber, DI - direct injection, FI - fuel injection.

(2) Prototype engines - not yet ready for production.

(1) NA - naturally aspirated, TC - turbocharged, PC - pre-chamber, DI - direct injection, FI - fuel injection.

(2) Prototype engines - not yet ready for production.

Table 9-23. Vehicle Emissions Test Data for Advanced Alternative Engines

Engine Type	Developer	Engine	Combustor	Vehicle	Inertia Weight, lb	Fuel	Emissions, g/mi			
							HC	CO	NO _x	
Brayton	Chrysler	6th generation	Diffusion flame, single can	Plymouth	4500	Diesel	0.68	3.6	2.8	
	Chrysler	6th generation	Premixed, torch ignition	Plymouth	4000	Gasoline	2.5	8.6	0.44	
	VW	GT-70	Toroidal	Microbus		JP-4	0.5	4.9	2.6	
	VW	GT-70	Toroidal	VW 1600		Gasoline	0.35	4.5	2.2	
	VW	GT-150	Diffusion flame, single can	RO-80	3700		<0.4	<3.4	>2.0	
	VW	GT-150	Premixed, single stage	RO-80	3700		<0.4	<3.4	>1, <2	
	GM	GT-225	Diffusion flame, single can	Impala	5000	Kerosene	0.2	2.8	6.0	
	GM	GT-225	Prechamber, premixed prevaporized	Impala	5000	Diesel	0.18	2.0	0.38	
	Williams	WR26	Toroidal	Hornet	3000	Gasoline	0.27	6.8	2.9	

Table 9-23. Vehicle Emissions Test Data for Advanced Alternative Engines
(Continuation 1)

Engine Type	Developer	Engine	Combustor	Vehicle	Inertia Weight, lb	Fuel	Emissions, g/mi		
							HC	CO	NO _x
Stirling	Ford	4-215	Diffusion flame, single can	Torino	4500	Gasoline	0.58	2.9	0.56
	Ford ⁽¹⁾	4-215	Diffusion flame, single can			Gasoline	0.04	2.0	0.39
	Ford ⁽²⁾	4-215	Diffusion flame, single can			Gasoline	0.10	0.3	0.17

(1) Projected emissions based on engine dynamometer test data.

(2) Projected emissions based on combustor rig test data.

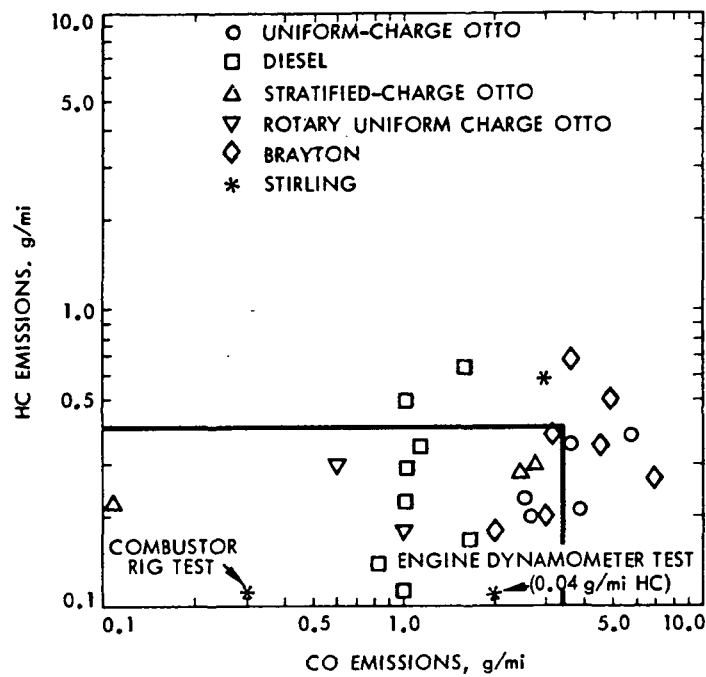


Figure 9-45. Urban Driving Cycle Emissions (HC, CO) for Various Engines

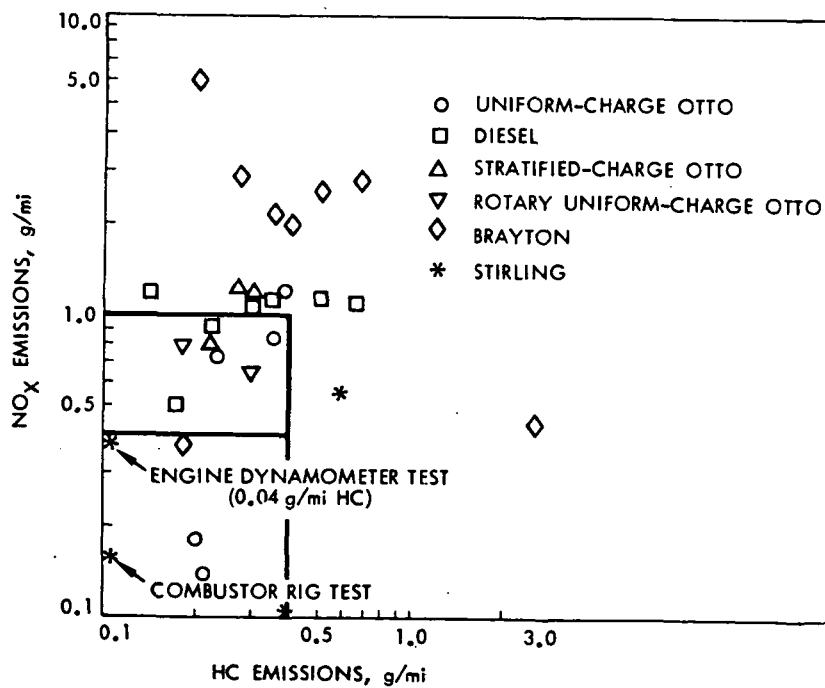


Figure 9-46. Urban Driving Cycle Emissions (NO_x, HC) for Various Engines

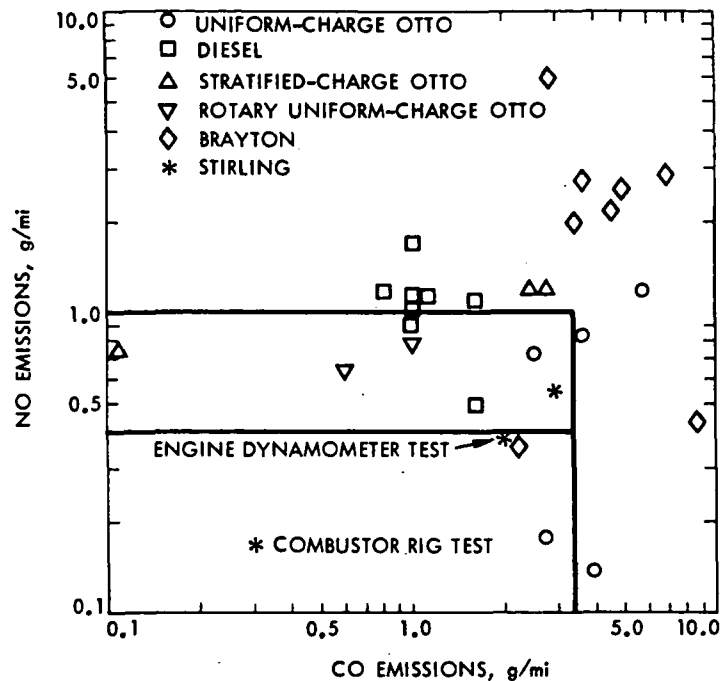


Figure 9-47. Urban Driving Cycle Emissions (NO_x , CO) for Various Engines

These emissions data indicate that there should be little difficulty meeting the 0.4 g/mi HC and 3.4 g/mi CO standards using conventional and advanced conventional engines. Some of the engines require the use of an oxidation catalyst, but catalyst technology is well developed and able to yield units with good conversion efficiency and adequate catalyst durability. Relative to the 1.0 g/mi NO_x standard, available data suggest that, with additional development, all conventional engines can meet this standard in vehicles of up to 4000 lb inertia weight. Most of the engines will require a three-way catalyst and/or exhaust gas recirculation (EGR) to reduce the NO_x emissions from current levels. The most difficult problem is with the diesel engine in the larger vehicles where the catalytic approach to reducing NO_x cannot be used and the presence of particulates in the exhaust for some operating conditions could lead to difficulties (durability of internal engine components) in using EGR.

Questions concerning whether or not the conventional type engines can be developed to meet the 0.4 g/mi NO_x standard and, if so, what fuel economy penalty is associated with reducing the NO_x emissions from 1.0 g/mi to 0.4 g/mi are difficult to answer based on available data. Reducing the NO_x emissions below 1.0 g/mi will be more difficult in the heavier vehicles. Present data indicate that for small vehicles (2250 lb inertia weight or less), the 0.4 g/mi NO_x standard can be met with conventional gasoline engines using a three-way catalyst and/or EGR. The fuel economy penalty at this vehicle weight should not be

significant. Reducing NO_x emissions in stratified-charge and diesel engines which operate at overall very lean conditions requires the use of increasing amounts of EGR, which has been demonstrated to result in significant fuel economy penalties (Reference 9-4). Hence for those engines the fuel economy penalty associated with NO_x emissions significantly below 1.0 g/mi are likely to be significant for cars heavier than 3000 lb inertia weight.

The limited vehicle emissions data for vehicles with Stirling engines indicate that there should be little difficulty in meeting the Federal legislated emissions standards. Although Stirling engines have not yet demonstrated their ability to meet the 0.4 g/mi NO_x standard in vehicle tests, projections based on engine dynamometer tests and combustor rig tests do indicate that these levels should be met. In fact, Stirling engines should have the cleanest exhaust of all engines being compared and should be able to meet the most stringent emissions requirements (0.4 g/mi HC, 3.4 g/mi CO, 0.4 g/mi NO_x) with relatively little additional development.

Steady-state emissions from Brayton engines are very low, but the effects of transient operation have yielded relatively high emissions levels in most vehicle tests. Using an advanced premixed, prevaporized combustor, General Motors has reported getting (0.18 g/mi HC, 2.0 g/mi CO, 0.38 g/mi NO_x) in a vehicle test; however, vehicle driveability was considered poor. The available vehicle and combustor emissions data indicate that low emissions can be achieved with the Brayton engine, but that careful combustor design and engine control (especially during transient conditions) are required. It is expected that with additional development both the Federal legislated emissions standards and the 0.4 g/mi NO_x research goal can be met by Brayton-powered vehicles.

Unregulated emissions (sulfates, particulates, and odor) are of concern for engines which use diesel fuel or distillates and/or operate at overall very lean conditions or locally very rich conditions. Sulfates result from sulfur in the fuel and can be controlled by setting strict permissible fuel sulfur regulations. This is true for all of the engines which use fuels other than gasoline - for which the sulfur content is already very low. Particulates (primarily smoke) result from injecting more fuel into the cylinders or combustor than can be burned in the residence time available. Smoking is most likely to occur in direct-injected and/or stratified-charge engines, such as the diesel, PROCO, and Texaco TCCS engines, and in engines using the heavier fuels. Smoking can be controlled to some extent by proper tailoring of the fuel injection system and its controls and by proper maintenance. The most difficult of the unregulated emissions to describe and to control is odor, which is primarily the result of very lean combustion. Odor is likely to be a problem with the diesel, gas turbine, PROCO, and Texaco TCCS engines.

As discussed in Section 6, the fuel requirements for the various engines differ markedly, varying from the need for high quality, high octane fuel by spark-ignited, Otto cycle engines to a tolerance to low quality wide-cut distillate fuels by the Brayton and Stirling engines. The fuel requirements of the various engines are summarized in Table 9-24. As noted in the table, most of the new engines currently being developed can operate satisfactorily on much lower quality fuel than the gasolines currently being produced. The alternative fuel capability of the new engines will become particularly important when energy resources other than crude oil are used to produce automotive fuels.

All of the fuels - gasoline, diesel oil, alcohols, and distillates - can be produced from a number of alternative energy resources, but, as is shown in Table 9-25, the thermal efficiency with which a particular fuel is produced differs significantly for the various resources. In general, the automotive fuels can be produced most efficiently from crude oil. To some extent, this is undoubtedly due to the high level of development of the refining technology, but to a greater extent it is due to the favorable carbon-to-hydrogen ratio of the crude oil itself. The fuel conversion efficiencies for shale oil and coal are significantly lower than for crude oil and also show a greater variation from fuel to fuel. It is clear from Table 9-25 that engines which yield high fuel economy in a vehicle and also can use wide-cut distillate fuels will be particularly attractive from an overall energy efficiency (resource-to-wheels) point-of-view. Brayton, Stirling, and direct-injection, stratified-charge engines fall into the high-energy-use efficiency group using the alternative resources, but the conventional diesel engine would not because of its requirement for a relatively high quality fuel (cetane number in a narrow range). From this same point-of-view, the long-term use of alcohol fuels does not appear attractive unless the fuel is produced from a renewable energy resource, such as biomass, in which case the overall energy efficiency is not critical as long as the cost of the alcohol fuel is competitive with that of other fuels.

Materials technology plays an important role in determining the attractiveness and ultimate feasibility of various automotive engines for a number of reasons: (1) material property limitations set the maximum temperatures and thus the peak thermal efficiency which can be attained; (2) material cost (\$/lb) and availability can be important factors in determining economic viability; (3) material preparation and component fabrication must be such that automation is possible to permit mass production. Various aspects of material technology for metals and ceramics are discussed in Sections 7 and 8 as they impact the design and manufacture of automotive engines. A summary of material requirements for the various engines is given in Table 9-26. For each engine, major individual components are listed along with the maximum temperature the component would experience and the type of material

Table 9-24. Fuel Requirements for Various Automotive Engines

Engine Type	Fuel Requirement	Fuel
Uniform-charge Otto Honda CVCC Ford PROCO	High volatility; high octane number, 90-95	Gasoline, alcohols
Diesel	Cetane number 40-65	Diesel oil
Direct-injected stratified-charge Texaco TCCS C-W rotary	None ⁽¹⁾	Wide-cut distillate
Spark-assisted diesel		
Brayton	None	Wide-cut distillate
Stirling	None	Wide-cut distillate
⁽¹⁾ The fuel injection and ignition systems for these engines can be designed to use fuels with a wide range of characteristics, but the systems must be specially engineered with the characteristics of the fuel to be used in mind.		

Table 9-25. Fuel Conversion Efficiency for Various Fuels
from Different Energy Resources (References
9-61, 9-62, 9-63)

Resource	Fuel	Resource Conversion Efficiency, %
Crude oil	Gasoline	90
	Diesel oil	92
	Distillates	94
Shale oil	Gasoline	65
	Diesel oil	67
	Distillates	70
Coal	Gasoline	55
	Methanol	45
	Distillates	67
Biomass	Ethanol	35

from which the component would be fabricated. The working environment (contact media, type and variation of stress, and thermal loads) for the components in each engine are given in Table 9-27.

The marked differences in the material requirements for the various engines are evident from Tables 9-26 and 9-27. In general, those for the continuous combustion engines, such as the Brayton and Stirling, are much more demanding than those for the conventional Otto and diesel engines. The Brayton and Stirling engines require the use of high temperature superalloy and ceramic materials because some of their components are subjected to high temperature, combustion gases, and/or the hot working medium essentially continuously when the engine is operating. For conventional engines, on the other hand, the critical engine surfaces, such as the pistons, valves, and cylinders, are subjected to high heat loads only intermittently and/or can be cooled with relative ease. Thus cast iron, low alloy steels, and aluminum alloys can be used in the conventional engines and superalloys and ceramics are not required even though the peak combustion gas temperature is 3000-4000°F.

Most of the materials technology needed for the metal Brayton and Stirling engines is available, having been developed in connection with aircraft Brayton engines. The exception is the ceramic cellular material needed for the regenerator in the automotive Brayton engine and the air preheater in the Stirling engine. As discussed in Section 8.4, ceramic core material (MAS and AS) is now available which

Table 9-26. Summary of Material Requirements for Conventional and Advanced Automotive Engines

Engine	Metal Parts		Ceramic Parts	
	Component/Temp. (1)/Material		Component/Temp. (1)/Material	
Conventional				
Uniform charge, stratified charge, reciprocating and rotary	Piston/650/cast iron low alloy steel, or aluminum alloy Block/350/cast iron		None	
Brayton (metal)	Compressor/350/aluminum alloy impeller Shafts/stainless steel Combustor/1900/superalloy Turbine/1900/superalloy Housings/cast iron		Regenerator/1500/MAS or AS	
Brayton (ceramic)	Compressor/350/aluminum alloy Shafts/stainless steel Housings/cast iron		Turbine/2500/SiC or Si ₃ N ₄ Regenerator/2000/MAS or AS Combustor/2500/SiC liner	
Stirling (metal)	Combustor/1800/superalloy Heater head/1400/superalloy Cylinder block/stainless steel Pistons/1400/superalloy		Air preheater/2000/MAS or AS	

Table 9-26. Summary of Material Requirements for Conventional and Advanced Automotive Engines (Continuation 1)

Engine	Metal Parts		Ceramic Parts	
	Component/Temp. (1) /Material		Component/Temp. (1) /Material	
Stirling (ceramic)	Cylinder block/stainless steel		Combustor/2500/SiC	
	Cycle coolers/stainless steel		Heater head/2000/SiC or Si ₃ N ₄	
	Swashplate drive/cast iron		Pistons/2000/SiC or Si ₃ N ₄	
			Cycle regenerator/2000/SiC	
			Air preheater/2700/MAS, AS, or SiC	

(1) All temperatures given in °F.

Table 9-27. Summary of Working Environment for Materials in Conventional and Advanced Automotive Engines

Engine/Components	Environment	Type of Stress load	Time Variation
Conventional (Otto and Diesel cycle)			
Pistons	Combustion gases	High pressure gases and reciprocating inertia loads	Cyclic and dependent on power
Brayton			
Compressor	Air	Rotational inertia	Dependent on engine RPM
Combustor liner	Combustion gases	Moderate pressure gases	Thermal load depends on power
Turbine (stator)	Combustion gases	Moderate pressure gases	Thermal load depends on power
Turbine (rotor)	Combustion gases	Rotational inertia	Dependent on engine RPM and power
Regenerator	Air and combustion gases (alternately)	Moderate pressure gases	Cyclic and dependent on power

Table 9-27. Summary of Working Environment for Materials in
Conventional and Advanced Automotive Engines
(Continuation 1)

Engine/Components	Environment	Type of Stress load	Time Variation
Stirling			
Combustor	Combustor gases	Low pressure gases	Dependent on power
Heater head (tubes)	Combustion gases (outside), hydrogen (inside)	Low pressure, high pressure	Dependent on power
Pistons	Hydrogen	High pressure	Dependent on power
Cycle regenerator	Hydrogen	High pressure	Dependent on power
Air preheater	Air and combustion gases (alternately)	Low pressure	Dependent on power

seems satisfactory for the regenerator/air preheater components. Hence components and even complete metal Brayton and Stirling engines for testing and development can be fabricated using materials which are currently available. The present high cost of the high temperature materials and the manufacturing processes (e.g., investment casting) used to form some of the key engine components combine to make the cost of the Brayton and Stirling engines considerably higher than the conventional engines, whose production can be highly automated using relatively low cost, easily worked materials.

Material requirements for some of the key components in the ceramic Brayton and Stirling engines are very severe and satisfactory materials are not yet available to meet them. In the case of the ceramic Brayton engines, the most critical component is the ceramic turbine rotor, which not only is subjected to high temperature (2500°F), but also high stress and thermal shock. For the ceramic Stirling engine, the critical components are the heater head and the air preheater. The heater head is a tubular heat exchanger unit which is in contact with 2500°F combustion gases on one side and 2000°F high pressure hydrogen on the other side. Little work has been done to date to fabricate such a unit from ceramics. The air preheater for the ceramic Stirling engine is subject to higher temperature combustion gases than the regenerator in a ceramic gas turbine (2700°F compared with 2000°F) and presently there is some doubt as to whether the materials being developed for the Brayton regenerator will be suitable for the Stirling engine application. Considerable R&D work is needed in ceramic materials characterization and processing before components for either the ceramic Brayton or Stirling engine will be available for developmental testing and subsequently for production engineering.

9.10 COST AND MANUFACTURABILITY

Comparisons of the projected costs of various alternative automotive engines with that of the conventional gasoline engine are both difficult and uncertain for several reasons. First, the projected cost of an alternative engine, such as the Brayton or Stirling, is highly dependent on a number of assumptions which must be made concerning material and labor costs and capital investments for facilities and machinery. The validity of the assumptions is very uncertain, especially for those engine components which utilize advanced materials and fabrication techniques for which there is no prior experience in a mass production industry such as the automobile industry. Second, none of the advanced engines has reached the point of production engineering - at which stage manufacturability and cost-effective design are emphasized. Third, the unit cost of producing the baseline conventional gasoline engine is not known with much certainty outside the automobile industry, making it difficult to set the reference engine cost. Selling prices of replacement gasoline and diesel engines are available and dealer price differentials between various engine options are known, but the relationship between those prices and the production costs are uncertain.

The engine cost results obtained in Section 7 are summarized in Table 9-28. Results are given for the variable cost, selling price, price difference, and price increase factor for the various automotive engines. The baseline engine is taken to be the conventional carbureted gasoline engine with an oxidation (two-way) catalyst. Cost projections are given in Table 9-28 for both 100 and 150 hp engines for each type. Extrapolation of the results to smaller 50-60 hp engines is probably not valid because the cost of the 100 hp engine was itself already inferred from that of the 150 hp engine and thus was not based on a complete new costing of a totally different (smaller block) engine.

The Brayton and Stirling engine results given in Table 9-28 are limited to engines using metallic components except for a ceramic core regenerator or air preheater. Sufficient information on manufacturing processes with structural ceramics is not yet available to permit the costing of ceramic engine components such as a turbine wheel or Stirling engine heater head.

The engine cost projections given in Table 9-28 indicate that both the Brayton and Stirling engines would have higher cost than the conventional gasoline engine at the same horsepower. The projected price increase factor would be about 1.4 for the Brayton engine and 1.55 for the Stirling engine. In the case of the Brayton engine, the price increase factor will be slightly lower than 1.4 if the engines are compared on the basis of equivalent performance in a vehicle rather than equal horsepower. For example, if a 125 hp gasoline engine is compared with a 100 hp FT Brayton engine, the estimated price increase factor would be 1.33 rather than 1.4. Further reductions in the price increase factor for the Brayton engine could result from the use of the single-shaft rather than the two-shaft configuration and the use of ceramics instead of superalloys for the turbine and combustor liner components. The magnitudes of the resultant cost reductions are very uncertain at the present time. Even without the possible cost reductions for the Brayton engine, the estimated price difference between the Brayton engine and the conventional gasoline engine is only \$500-\$600 (1977 dollars) for a full-sized car. This difference is only about 10 percent of the current selling price of a car of that size and does not seem to preclude, on an economic basis, the introduction of the gas turbine, especially if other characteristics of the engine, such as performance and smoothness/quietness, are found to be attractive by automobile buyers. The cost difference between the Brayton and diesel engine is estimated to be \$300-\$400.

As indicated in Table 9-28, the cost difference between the Stirling engine and the conventional gasoline engine is estimated to be about \$800 (1977 dollars) for engines in the 100-150 hp size range. This price difference is not likely to be significantly affected by scaling or ceramic use considerations as was the case for the Brayton engine. The relatively large cost differential for the Stirling engine will thus need to be justified by improved fuel economy, lower emissions, and greater fuel versatility. In that sense, the Stirling engine situation is comparable to that of the diesel engine, which is

Table 9-28. Summary of Cost Projections for
Various Automotive Engines

Engine	Variable Cost, \$	Selling Price, \$	Price Differential, \$	Price Increase Factor
Gasoline- carbureted with catalyst				
100 hp	400	1403	0	1.0
150 hp	485	1555	0	1.0
Gasoline - fuel injected with 3-way catalyst				
100 hp	420	1474	71	1.05
150 hp	500	1617	62	1.04
Diesel - turbocharged				
150 hp	606	1786	169	1.13
Brayton (metallic) - 2 shaft				
100 hp	651	2002	599	1.43
150 hp	711	2108	553	1.36
Stirling (metallic)				
100 hp	770	2208	805	1.57
150 hp	875	2399	844	1.54

currently being introduced at a relatively large cost premium almost completely because of its higher fuel economy. Even though the initial cost picture for the Stirling engine appears to be less favorable than that for the gas turbine, it does not seem to preclude its eventual introduction if its excellent fuel economy potential can be realized.

9.11 PRESENT STATUS AND PROJECTED AVAILABILITY

The present status of the technology and the projected availability in the marketplace of various alternatives to the conventional gasoline engine for use in passenger cars are compared in this section. The present status will be identified in terms of one of the following categories: (1) production for market, (2) prototype testing of the engine in vehicles, (3) laboratory testing of individual engines, (4) engineering studies (paper) of engine configurations, and (5) materials R&D. The availability of the engine will be ascribed to one of three stages: (1) required component technology available, (2) prototype development in progress, and (3) limited production for market possible. A date (year) or period (years) will be assigned to each stage of availability for the various engines. It will be assumed that for each engine systematic and well-funded R&D and engineering/test programs are being planned and/or will be undertaken starting at the present time (mid-1978). In addition, it will be assumed that ultimate decisions relative to production will be made based solely on the adequacy of the technology and fabrication techniques and not the relative economic viability of the various engines. In other words, the development of each engine type would be a separate program independent of the existence and progress in other such programs. With limited development funds available, competing R&D programs are not usually conducted in that way, but to assume otherwise makes technology forecasting even more difficult.

Using the category definitions and program administrative philosophy indicated in the previous paragraph, a summary of the present status and projected availability of various alternative automotive engines is given in Table 9-29. There is clearly considerable speculation and uncertainty in preparing such a broad forecast. Probably the most speculative input is the assumption that R&D work in ceramics will lead in about 5 years to the materials technology required for the ceramic Brayton and Stirling engine. In nearly all other areas, most of the technology base is presently available for the design and fabrication of components and laboratory engines. The remaining work to be done involves extensive engineering design and testing of prototype components/engines and manufacturing/production oriented activities leading to mass production.

The forecast given in Table 9-29 indicates that by 1985 a number of alternative automotive engines can be developed to a production-ready status. Of course, which of these engines would warrant large-scale production, based on demonstrated fuel economy and cost, will depend on the outcome of the extensive development programs

Table 9-29. Summary of the Present Status and Projected Availability of Various Alternative Automotive Engines

Engine	Present Status	Availability of Required Component Technology	Prototype Development	Production for Market
Diesel	Production	-	-	-
Diesel (turbocharged)	Prototype	Now	Now	1980
Stratified charge (direct injection)				
Reciprocating (TCCS)	Laboratory testing	Now	1980-85	1985
Rotary (C-W)	Laboratory testing	Now	1980-85	1985
Brayton (metal)				
2-shaft	Laboratory testing	Now	1980-85	1985
Single-shaft	Engineering studies	Now (except for CVT)	1980-85	1985
Brayton (ceramic)				
2-shaft	Materials R&D	1985	1985-90	1990
Single-shaft	Materials R&D	1985	1985-90	1990
Stirling (metal)	Laboratory testing	Now	1980-85	1985
Stirling (ceramic)	Materials R&D	1985	1985-90	1990

between now and 1985. Undoubtedly, well before 1985, the turbocharged diesel will be marketed in significant quantities and its success⁽¹⁾ in passenger cars may determine the relative attractiveness of the other alternative engines. Engines using critical ceramic components are not projected to be available before 1990 at the earliest.

9.12 CONCLUSIONS

Based on the comparisons of the potential of the various engine systems, the following general conclusions can be drawn relative to their fuel economy, emissions, multifuel capability, and cost:

FUEL ECONOMY

- Projected fuel economies of Stirling vehicles are up to 40 percent better than 1978 baseline (1.0 g/mi NO_x) and up to 40 percent better than the projected 1985 baseline (0.4 g/mi NO_x).
- Projected fuel economies of Brayton vehicles are up to 30 percent better than 1978 baseline (1.0 g/mi NO_x) and up to 30 percent better than the projected 1985 baseline (0.4 g/mi NO_x) in full-sized vehicles, but offer little advantage in small-sized vehicles.
- Ceramic materials are needed to show any significant fuel economy advantage for Brayton vehicles.
- Baseline vehicle fuel economies will likely continue their improvement with new lighter weight engine designs and continued work on turbocharging.

EMISSIONS

- Advanced continuous combustion power systems (Brayton and Stirling) should meet the emissions research goal (0.4 g/mi NO_x) and the currently legislated emissions standards in both small and large vehicles.
- Conventional UC Otto vehicles with three-way catalyst emissions control should meet the legislated emissions standards (1.0 g/mi NO_x) with no fuel economy penalty. These vehicles should meet the 0.4 g/mi NO_x research goal in small cars with little fuel economy penalty.

(1) Based on customer acceptance, durability, cost, and environmental impact

- Diesel vehicles should meet the legislated emissions standards. This probably requires the use of advanced emissions control techniques on larger vehicles.
- Unregulated emissions (particulates, odor, and noise) will present problems for diesel vehicles if new emissions regulations are passed.

MULTIFUEL CAPABILITY

- Advanced continuous combustion power systems (Brayton and Stirling) should perform well using a wide range of fuels.
- Some advanced conventional engines under development have demonstrated limited multifuel capability.

COST

- The estimated price difference between the Brayton engine and the conventional gasoline engine is \$500-\$600 (1977 dollars) for a full-sized car.
- The cost difference between the Stirling and the conventional gasoline engine is estimated to be about \$800 (1977 dollars) for engines in the 100-150 hp size range.

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